Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

5th Edition

Yunus A. Cengel & Afshin J. Ghajar

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Chapter 11
HEAT EXCHANGERS

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Types of Heat Exchangers

11-1C Heat exchangers are classified according to the flow type as parallel flow, counter flow, and cross-flow arrangement. In parallel flow, both the hot and cold fluids enter the heat exchanger at the same end and move in the same direction. In counter-flow, the hot and cold fluids enter the heat exchanger at opposite ends and flow in opposite direction. In cross-flow, the hot and cold fluid streams move perpendicular to each other.

11-2C A heat exchanger is classified as being compact if $\beta > 700\ m^2/m^3$ or ($200\ ft^2/ft^3$) where β is the ratio of the heat transfer surface area to its volume which is called the area density. The area density for double-pipe heat exchanger can not be in the order of 700. Therefore, it can not be classified as a compact heat exchanger.

11-3C Regenerative heat exchanger involves the alternate passage of the hot and cold fluid streams through the same flow area. The static type regenerative heat exchanger is basically a porous mass which has a large heat storage capacity, such as a ceramic wire mash. Hot and cold fluids flow through this porous mass alternately. Heat is transferred from the hot fluid to the matrix of the regenerator during the flow of the hot fluid and from the matrix to the cold fluid. Thus the matrix serves as a temporary heat storage medium. The dynamic type regenerator involves a rotating drum and continuous flow of the hot and cold fluid through different portions of the drum so that any portion of the drum passes periodically through the hot stream, storing heat and then through the cold stream, rejecting this stored heat. Again the drum serves as the medium to transport the heat from the hot to the cold fluid stream.

11-4C In the shell and tube exchangers, baffles are commonly placed in the shell to force the shell side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes. Baffles disrupt the flow of fluid, and an increased pumping power will be needed to maintain flow. On the other hand, baffles eliminate dead spots and increase heat transfer rate.

11-5C Using six-tube passes in a shell and tube heat exchanger increases the heat transfer surface area, and the rate of heat transfer increases. But it also increases the manufacturing costs.

11-6C Using so many tubes increases the heat transfer surface area which in turn increases the rate of heat transfer.

11-7C In counter-flow heat exchangers, the hot and the cold fluids move parallel to each other but both enter the heat exchanger at opposite ends and flow in opposite direction. In cross-flow heat exchangers, the two fluids usually move perpendicular to each other. The cross-flow is said to be unmixed when the plate fins force the fluid to flow through a particular interfin spacing and prevent it from moving in the transverse direction. When the fluid is free to move in the transverse direction, the cross-flow is said to be mixed.

The Overall Heat Transfer Coefficient

11-8C Heat is first transferred from the hot liquid to the wall by convection, through the wall by conduction and from the wall to the cold liquid again by convection.

11-9C When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, which is usually the case, the thermal resistance of the tube is negligible.

11-10C The heat transfer surface areas are $A_i = \pi D_1 L$ and $A_o = \pi D_2 L$. When the thickness of inner tube is small, it is reasonable to assume $A_i \cong A_o \cong A_s$.

11-11C The effect of fouling on a heat transfer is represented by a fouling factor R_f. Its effect on the heat transfer coefficient is accounted for by introducing a thermal resistance R_f / A_s. The fouling increases with increasing temperature and decreasing velocity.

11-12C None.

11-13C When one of the convection coefficients is much smaller than the other $h_i << h_o$, and $A_i \approx A_0 \approx A_s$. Then we have ($1/h_i >> 1/h_o$) and thus $U_i = U_0 = U \cong h_i$.

11-14C The most common type of fouling is the precipitation of solid deposits in a fluid on the heat transfer surfaces. Another form of fouling is corrosion and other chemical fouling. Heat exchangers may also be fouled by the growth of algae in warm fluids. This type of fouling is called the biological fouling. Fouling represents additional resistance to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease, and the pressure drop to increase.

11-15C When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, the thermal resistance of the tube is negligible and the inner and the outer surfaces of the tube are almost identical ($A_i \cong A_o \cong A_s$). Then the overall heat transfer coefficient of a heat exchanger can be determined to from $U = (1/h_i + 1/h_o)^{-1}$

11-16E The overall heat transfer coefficients based on the outer and inner surfaces for a heat exchanger are to be determined.

Assumptions **1** Steady operating condition exists. **2** Thermal properties are constant.

Properties The conductivity of the tube material is given to be 0.5 Btu/hr·ft·°F.

Analysis The overall heat transfer coefficient based on the outer surface is

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{1}{2\pi kL}\ln(D_o / D_i) + \frac{1}{h_o A_o}$$

$$\frac{1}{U_o} = \frac{A_o}{h_i A_i} + \frac{A_o}{2\pi kL}\ln\left(\frac{D_o}{D_i}\right) + \frac{A_o}{h_o A_o} = \frac{1}{h_i}\frac{D_o}{D_i} + \frac{D_o}{2k}\ln\left(\frac{D_o}{D_i}\right) + \frac{1}{h_o}$$

Thus

$$U_o = \left[\frac{1}{h_i}\frac{D_o}{D_i} + \frac{D_o}{2k}\ln\left(\frac{D_o}{D_i}\right) + \frac{1}{h_o}\right]^{-1}$$

$$= \left[\left(\frac{1}{50}\right)\left(\frac{3}{2}\right) + \frac{3/12}{2(0.5)}\ln\left(\frac{3}{2}\right) + \frac{1}{10}\right]^{-1} \text{ Btu/hr}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

$$= \mathbf{4.32\ Btu/hr\cdot ft^2\cdot{}^\circ F}$$

The overall heat transfer coefficient based on the inner surface is

$$\frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{1}{2\pi kL}\ln(D_o / D_i) + \frac{1}{h_o A_o}$$

$$\frac{1}{U_i} = \frac{A_i}{h_i A_i} + \frac{A_i}{2\pi kL}\ln\left(\frac{D_o}{D_i}\right) + \frac{A_i}{h_o A_o} = \frac{1}{h_i} + \frac{D_i}{2k}\ln\left(\frac{D_o}{D_i}\right) + \frac{1}{h_o}\frac{D_i}{D_o}$$

Thus

$$U_i = \left[\frac{1}{h_i} + \frac{D_i}{2k}\ln\left(\frac{D_o}{D_i}\right) + \frac{1}{h_o}\frac{D_i}{D_o}\right]^{-1}$$

$$= \left[\left(\frac{1}{50}\right) + \frac{2/12}{2(0.5)}\ln\left(\frac{3}{2}\right) + \frac{1}{10}\left(\frac{2}{3}\right)\right]^{-1} \text{ Btu/hr}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

$$= \mathbf{6.48\ Btu/hr\cdot ft^2\cdot{}^\circ F}$$

Discussion The two overall heat transfer coefficients differ significantly with U_i larger than U_o by a factor of 1.5. The overall heat transfer coefficient ratio can be expressed as

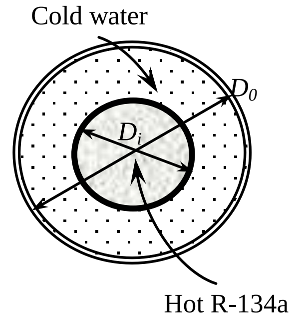
$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} \quad \rightarrow \quad \frac{U_i}{U_o} = \frac{A_o}{A_i} = \frac{D_o}{D_i} = 1.5$$

11-17 Refrigerant-134a is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions **1** The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and refrigerant-134a flow are fully developed. **3** Properties of the water and refrigerant-134a are constant.

Properties The properties of water at 20°C are (Table A-9)

$$\rho = 998\ \text{kg/m}^3$$
$$\nu = \mu/\rho = 1.004\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.598\ \text{W/m°C}$$
$$\text{Pr} = 7.01$$



Analysis The hydraulic diameter for annular space is

$$D_h = D_o - D_i = 0.025 - 0.01 = 0.015\ \text{m}$$

The average velocity of water in the tube and the Reynolds number are

$$V_{avg} = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho\left(\pi \frac{D_o^2 - D_i^2}{4}\right)} = \frac{0.3\ \text{kg/s}}{(998\ \text{kg/m}^3)\left(\pi \frac{(0.025\ \text{m})^2 - (0.01\ \text{m})^2}{4}\right)} = 0.729\ \text{m/s}$$

$$\text{Re} = \frac{V_{avg} D_h}{\nu} = \frac{(0.729\ \text{m/s})(0.015\ \text{m})}{1.004\times10^{-6}\ \text{m}^2/\text{s}} = 10{,}890$$

which is greater than 4000. Therefore flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}890)^{0.8}(7.01)^{0.4} = 85.0$$

and

$$h_o = \frac{k}{D_h} Nu = \frac{0.598\ \text{W/m.°C}}{0.015\ \text{m}}(85.0) = 3390\ \text{W/m}^2.\text{°C}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{4100\ \text{W/m}^2.\text{°C}} + \frac{1}{3390\ \text{W/m}^2.\text{°C}}} = \mathbf{1856\ W/m^2.°C}$$

Discussion This problem can also be solved using Gnielinski equation. First the friction factor is determined from the first Petukhov equation.

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = [0.790\ln(10{,}890) - 1.64]^{-2} = 0.03074$$

$$Nu = \frac{hD_h}{k} = \frac{(f/8)(\text{Re}-1000)\,\text{Pr}}{1 + 12.7(f/8)^{0.5}(\text{Pr}^{2/3} - 1)} = \frac{(0.03074/8)(10{,}890 - 1000)(7.01)}{1 + 12.7(0.03074/8)^{0.5}(7.01^{2/3} - 1)} = 86.04$$

$$h_o = \frac{k}{D_h} Nu = \frac{0.598\ \text{W/m.°C}}{0.015\ \text{m}}(86.04) = 3430\ \text{W/m}^2.\text{°C}$$

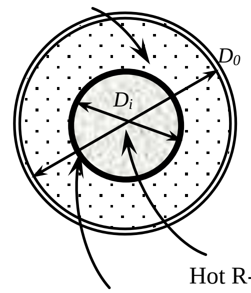
$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{4100\ \text{W/m}^2.\text{°C}} + \frac{1}{3430\ \text{W/m}^2.\text{°C}}} = \mathbf{1868\ W/m^2.°C}$$

The result is very close to that obtained by using the modified Colburn equation for the Nusselt number. Therefore, different heat transfer correlations can be used to solve for the heat transfer coefficient.

11-18 Refrigerant-134a is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions **1** The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and refrigerant-134a flows are fully developed. **3** Properties of the water and refrigerant-134a are constant. **4** The limestone layer can be treated as a plain layer since its thickness is very small relative to its diameter.

Properties The properties of water at 20°C are (Table A-9)



$$\rho = 998\ \text{kg/m}^3$$
$$\nu = \mu/\rho = 1.004\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.598\ \text{W/m°C}$$
$$\text{Pr} = 7.01$$

Analysis The hydraulic diameter for annular space is

$$D_h = D_o - D_i = 0.025 - 0.01 = 0.015\ \text{m}$$

The average velocity of water in the tube and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho\left(\pi\dfrac{D_o^{\,2} - D_i^{\,2}}{4}\right)} = \frac{0.3\ \text{kg/s}}{(998\ \text{kg/m}^3)\left(\pi\dfrac{(0.025\ \text{m})^2 - (0.01\ \text{m})^2}{4}\right)} = 0.729\ \text{m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.729\ \text{m/s})(0.015\ \text{m})}{1.004\times10^{-6}\ \text{m}^2/\text{s}} = 10{,}890$$

which is greater than 10,000. Therefore flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}890)^{0.8}(7.01)^{0.4} = 85.0$$

and

$$h_o = \frac{k}{D_h}Nu = \frac{0.598\ \text{W/m.°C}}{0.015\ \text{m}}(85.0) = 3390\ \text{W/m}^2.\text{°C}$$

Disregarding the curvature effects, the overall heat transfer coefficient is determined to be

$$U = \frac{1}{\dfrac{1}{h_i} + \left(\dfrac{L}{k}\right)_{\text{limeston}} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{4100\ \text{W/m}^2.\text{°C}} + \dfrac{0.002\ \text{m}}{1.3\ \text{W/m°C}} + \dfrac{1}{3390\ \text{W/m}^2.\text{°C}}} = \mathbf{481\,W/m^2.°C}$$

11-19 Prob. 11-18 is reconsidered. The overall heat transfer coefficient as a function of the limestone thickness is to be plotted.

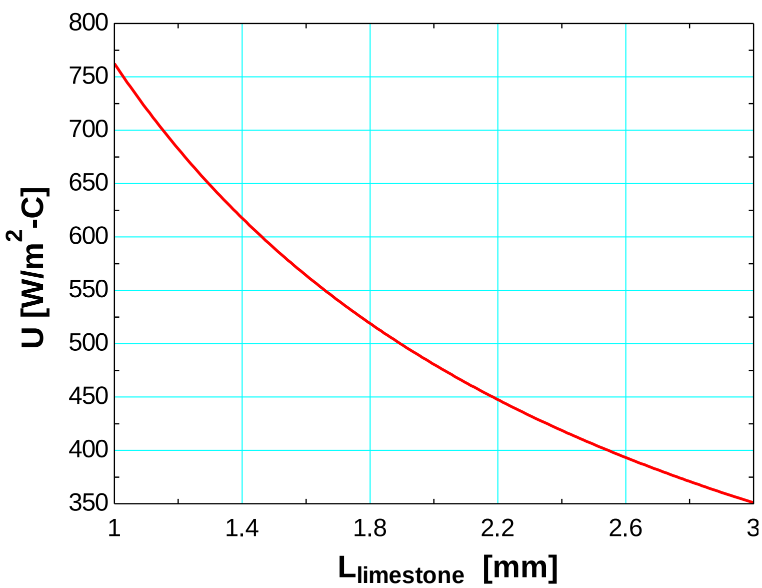
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
D_i=0.010 [m]
D_o=0.025 [m]
T_w=20 [C]
h_i=4100 [W/m^2-C]
m_dot=0.3 [kg/s]
L_limestone=2 [mm]
k_limestone=1.3 [W/m-C]

"PROPERTIES"
k=conductivity(Water, T=T_w, P=100)
Pr=Prandtl(Water, T=T_w, P=100)
rho=density(Water, T=T_w, P=100)
mu=viscosity(Water, T=T_w, P=100)
nu=mu/rho

"ANALYSIS"
D_h=D_o-D_i
Vel=m_dot/(rho*A_c)
A_c=pi*(D_o^2-D_i^2)/4
Re=(Vel*D_h)/nu
"Re is calculated to be greater than 10,000. Therefore, the flow is turbulent."
Nusselt=0.023*Re^0.8*Pr^0.4
h_o=k/D_h*Nusselt
U=1/(1/h_i+(L_limestone*Convert(mm, m))/k_limestone+1/h_o)
```

$L_{limestone}$ [mm]	U [W/m^2-C]
1	762.4
1.1	720.2
1.2	682.4
1.3	648.3
1.4	617.5
1.5	589.5
1.6	564
1.7	540.5
1.8	518.9
1.9	499
2	480.6
2.1	463.4
2.2	447.5
2.3	432.6
2.4	418.7
2.5	405.6
2.6	393.3
2.7	381.8
2.8	370.9
2.9	360.6
3	350.9

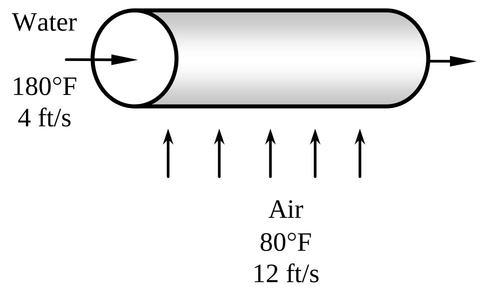


11-20E Water is cooled by air in a cross-flow heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions **1** The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. **2** Both the water and air flow are fully developed. **3** Properties of the water and air are constant.

Properties The properties of water at 180°F are (Table A-9E)

$$k = 0.388\ \text{Btu/h.ft.°F}$$
$$\nu = 3.825\times10^{-6}\ \text{ft}^2/\text{s}$$
$$\text{Pr} = 2.15$$



The properties of air at 80°F are (Table A-15E)

$$k = 0.01481\ \text{Btu/h.ft.°F}$$
$$\nu = 1.697\times10^{-4}\ \text{ft}^2/\text{s}$$
$$\text{Pr} = 0.7290$$

Analysis The overall heat transfer coefficient can be determined from

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o}$$

The Reynolds number of water is

$$\text{Re} = \frac{V_{avg}D_h}{\nu} = \frac{(4\ \text{ft/s})[0.75/12\ \text{ft}]}{3.825\times10^{-6}\ \text{ft}^2/\text{s}} = 65{,}360$$

which is greater than 10,000. Therefore the flow of water is turbulent. Assuming the flow to be fully developed, the Nusselt number is determined from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(65{,}360)^{0.8}(2.15)^{0.4} = 222$$

and $$h_i = \frac{k}{D_h}Nu = \frac{0.388\ \text{Btu/h.ft.°F}}{0.75/12\ \text{ft}}(222) = 1378\ \text{Btu/h.ft}^2\text{.°F}$$

The Reynolds number of air is

$$\text{Re} = \frac{VD}{\nu} = \frac{(12\ \text{ft/s})[3/(4\times12)\ \text{ft}]}{1.697\times10^{-4}\ \text{ft}^2/\text{s}} = 4420$$

The flow of air is across the cylinder. The proper relation for Nusselt number in this case is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(4420)^{0.5}(0.7290)^{1/3}}{\left[1+(0.4/0.7290)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4420}{282{,}000}\right)^{5/8}\right]^{4/5} = 34.86$$

and $$h_o = \frac{k}{D}Nu = \frac{0.01481\ \text{Btu/h.ft.°F}}{0.75/12\ \text{ft}}(34.86) = 8.26\ \text{Btu/h.ft}^2\text{.°F}$$

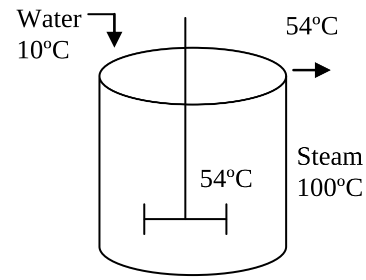
Then the overall heat transfer coefficient becomes

$$U = \frac{1}{\dfrac{1}{h_i}+\dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{1378\ \text{Btu/h.ft}^2\text{.°F}} + \dfrac{1}{8.26\ \text{Btu/h.ft}^2\text{.°F}}} = \mathbf{8.21\ Btu/h.ft^2.°F}$$

11-21 A water stream is heated by a jacketted-agitated vessel, fitted with a turbine agitator. The mass flow rate of water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The properties of water at 54°C are (Table A-9)



$$k = 0.648 \text{ W/m.°C}$$
$$\rho = 985.8 \text{ kg/m}^3$$
$$\mu = 0.513\times10^{-3} \text{ kg/m}\cdot\text{s}$$
$$\Pr = 3.31$$

The specific heat of water at the average temperature of (10+54)/2=32°C is 4178 J/kg.°C (Table A-9)

Analysis We first determine the heat transfer coefficient on the inner wall of the vessel

$$\text{Re} = \frac{\dot{n}D_a^2\rho}{\mu} = \frac{(60/60 \text{ s}^{-1})(0.2 \text{ m})^2(985.8 \text{ kg/m}^3)}{0.513\times10^{-3} \text{ kg/m}\cdot\text{s}} = 76{,}865$$

$$Nu = 0.76\,\text{Re}^{2/3}\,\Pr^{1/3} = 0.76(76{,}865)^{2/3}(3.31)^{1/3} = 2048$$

$$h_j = \frac{k}{D_t}Nu = \frac{0.648 \text{ W/m.°C}}{0.6 \text{ m}}(2048) = 2211 \text{ W/m}^2\text{.°C}$$

The heat transfer coefficient on the outer side is determined as follows

$$h_o = 13{,}100(T_g - T_w)^{-0.25} = 13{,}100(100 - T_w)^{-0.25}$$

$$h_o(T_g - T_w) = h_j(T_w - 54)$$
$$13{,}100(100 - T_w)^{-0.25}(100 - T_w) = 2211(T_w - 54)$$
$$13{,}100(100 - T_w)^{0.75} = 2211(T_w - 54)$$
$$\rightarrow T_w = 89.2°\text{C}$$

$$h_o = 13{,}100(100 - T_w)^{-0.25} = 13{,}100(100 - 89.2)^{-0.25} = 7226 \text{ W/m}^2\cdot\text{C}$$

Neglecting the wall resistance and the thickness of the wall, the overall heat transfer coefficient can be written as

$$U = \left(\frac{1}{h_j} + \frac{1}{h_o}\right)^{-1} = \left(\frac{1}{2211} + \frac{1}{7226}\right)^{-1} = 1694 \text{ W/m}^2\cdot\text{C}$$

From an energy balance

$$[\dot{m}c(T_{out} - T_{in})]_{\text{water}} = UA\Delta T$$
$$\dot{m}_w(4178)(54 - 10) = (1694)(\pi\times0.6\times0.6)(100 - 54)$$
$$\dot{m}_w = 0.479 \text{ kg/s} = \mathbf{1725 kg/h}$$

11-22E The overall heat transfer coefficient of a heat exchanger and the percentage change in the overall heat transfer coefficient due to scale built-up are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat transfer coefficients and the fouling factors are constant and uniform.

Analysis When operating at design and clean conditions, the overall heat transfer coefficient is given as

$$U_{\text{w/o scale}} = 50\ \text{Btu/hr}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

(*a*) After a period of use, the overall heat transfer coefficient due to the scale built-up is

$$\begin{aligned}\frac{1}{U_{\text{w/ scale}}} &= \frac{1}{U_{\text{w/o scale}}} + R_f \\ &= \frac{1}{50\ \text{Btu/hr}\cdot\text{ft}^2\cdot{}^\circ\text{F}} + 0.002\ \text{hr}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu} \\ &= 0.022\ \text{hr}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu}\end{aligned}$$

or

$$U_{\text{w/ scale}} = \mathbf{45.5\ Btu/hr\cdot ft^2\cdot{}^\circ F}$$

(*b*) The percentage change in the overall heat transfer coefficient due to the scale built-up is

$$\frac{U_{\text{w/o scale}} - U_{\text{w/ scale}}}{U_{\text{w/o scale}}}\times 100 = \frac{50-45.5}{50}\times 100 = \mathbf{9\%}$$

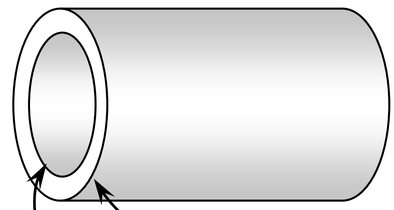
Discussion The scale built-up caused a 9% decrease in the overall heat transfer coefficient of the heat exchanger.

11-23 Water flows through the tubes in a boiler. The overall heat transfer coefficient of this boiler based on the inner surface area is to be determined.

Assumptions **1** Water flow is fully developed. **2** Properties of the water are constant.

Properties The properties of water at 110°C are (Table A-9)

$$\nu = \mu / \rho = 0.268\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.682\ \text{W/m}^2.\text{K}$$
$$\Pr = 1.58$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.5\ \text{m/s})(0.01\text{m})}{0.268\times10^{-6}\ \text{m}^2/\text{s}} = 130{,}600$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\Pr^{0.4} = 0.023(130{,}600)^{0.8}(1.58)^{0.4} = 341.9$$

and

$$h = \frac{k}{D_h} Nu = \frac{0.682\ \text{W/m.°C}}{0.01\,\text{m}}(341.9) = 23{,}320\ \text{W/m}^2.\text{°C}$$

The total resistance of this heat exchanger is then determined from

$$R = R_{total} = R_i + R_{wall} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{1}{h_o A_o}$$
$$= \frac{1}{(23{,}320\ \text{W/m}^2.\text{°C})[\pi(0.01\,\text{m})(5\,\text{m})]} + \frac{\ln(1.4/1)}{[2\pi(14.2\ \text{W/m.°C})(5\,\text{m})]} + \frac{1}{(8400\ \text{W/m}^2.\text{°C})[\pi(0.014\,\text{m})(5\,\text{m})]}$$
$$= 0.001569\text{°C/W}$$

and

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.001569\text{°C/W})[\pi(0.01\,\text{m})(5\,\text{m})]} = \mathbf{4057\ W/m^2.°C}$$

11-24 Water is flowing through the tubes in a boiler. The overall heat transfer coefficient of this boiler based on the inner surface area is to be determined.

Assumptions **1** Water flow is fully developed. **2** Properties of water are constant. **3** The heat transfer coefficient and the fouling factor are constant and uniform.

Properties The properties of water at 110°C are (Table A-9)

$$\nu = \mu / \rho = 0.268\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.682\ \text{W/m}^2.\text{K}$$
$$\text{Pr} = 1.58$$

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3.5\ \text{m/s})(0.01\text{m})}{0.268\times10^{-6}\ \text{m}^2/\text{s}} = 130{,}600$$

which is greater than 10,000. Therefore, the flow is turbulent. Assuming fully developed flow,

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(130{,}600)^{0.8}(1.58)^{0.4} = 341.9$$

and

$$h = \frac{k}{D_h} Nu = \frac{0.682\ \text{W/m.°C}}{0.01\,\text{m}}(341.9) = 23{,}320\ \text{W/m}^2.\text{°C}$$

The thermal resistance of heat exchanger with a fouling factor of $R_{f,i} = 0.0005\,\text{m}^2.\text{°C/W}$ is determined from

$$R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o / D_i)}{2\pi kL} + \frac{1}{h_o A_o}$$

$$R = \frac{1}{(23{,}320\ \text{W/m}^2.\text{°C})[\pi(0.01\,\text{m})(5\,\text{m})]} + \frac{0.0005\,\text{m}^2.\text{°C/W}}{[\pi(0.01\,\text{m})(5\,\text{m})]}$$
$$+ \frac{\ln(1.4/1)}{2\pi(14.2\ \text{W/m.°C})(5\,\text{m})} + \frac{1}{(8400\ \text{W/m}^2.\text{°C})[\pi(0.014\,\text{m})(5\,\text{m})]}$$
$$= 0.004752\text{°C/W}$$

Then,

$$R = \frac{1}{U_i A_i} \longrightarrow U_i = \frac{1}{RA_i} = \frac{1}{(0.004752\text{°C/W})[\pi(0.01\,\text{m})(5\,\text{m})]} = \mathbf{1340\,W/m^2.°C}$$

11-25 Prob. 11-24 is reconsidered. The overall heat transfer coefficient based on the inner surface as a function of fouling factor is to be plotted.

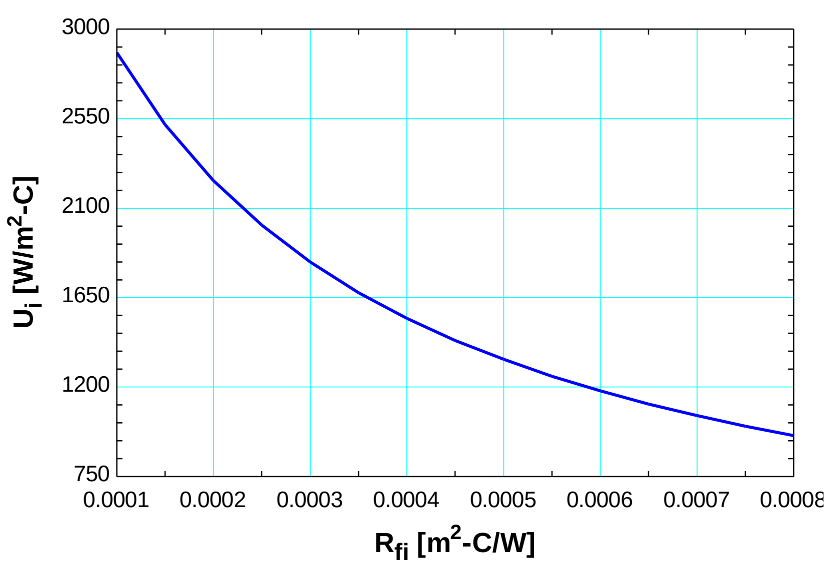
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_w=110 [C]
Vel=3.5 [m/s]
L=5 [m]
k_pipe=14.2 [W/m-C]
D_i=0.010 [m]
D_o=0.014 [m]
h_o=8400 [W/m^2-C]
R_f_i=0.0005 [m^2-C/W]

"PROPERTIES"
k=conductivity(Water, T=T_w, P=300)
Pr=Prandtl(Water, T=T_w, P=300)
rho=density(Water, T=T_w, P=300)
mu=viscosity(Water, T=T_w, P=300)
nu=mu/rho

"ANALYSIS"
Re=(Vel*D_i)/nu  "Re is calculated to be greater than 10,000. Therefore, the flow is turbulent."
Nusselt=0.023*Re^0.8*Pr^0.4
h_i=k/D_i*Nusselt
A_i=pi*D_i*L
A_o=pi*D_o*L
R=1/(h_i*A_i)+R_f_i/A_i+ln(D_o/D_i)/(2*pi*k_pipe*L)+1/(h_o*A_o)
U_i=1/(R*A_i)
```

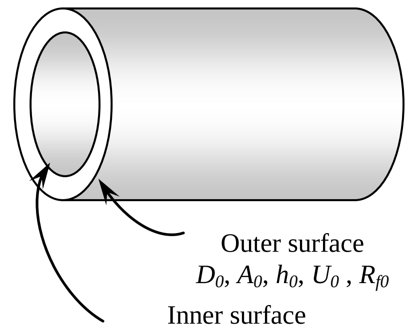
$R_{f,i}$ [m^2-C/W]	U_i [W/m^2-C]
0.0001	2882
0.00015	2519
0.0002	2238
0.00025	2012
0.0003	1828
0.00035	1675
0.0004	1546
0.00045	1435
0.0005	1339
0.00055	1255
0.0006	1181
0.00065	1115
0.0007	1056
0.00075	1003
0.0008	955.2



11-26 The heat transfer coefficients and the fouling factors on tube and shell side of a heat exchanger are given. The thermal resistance and the overall heat transfer coefficients based on the inner and outer areas are to be determined.

Assumptions **1** The heat transfer coefficients and the fouling factors are constant and uniform.

Analysis (*a*) The total thermal resistance of the heat exchanger per unit length is



$$R=\frac{1}{h_i A_i}+\frac{R_{fi}}{A_i}+\frac{\ln(D_o/D_i)}{2\pi kL}+\frac{R_{fo}}{A_o}+\frac{1}{h_o A_o}$$

$$R=\frac{1}{(700\text{ W/m}^2.°\text{C})[\pi(0.012\text{ m})(1\text{ m})]}+\frac{(0.0005\text{ m}^2.°\text{C/W})}{[\pi(0.012\text{ m})(1\text{ m})]}$$
$$+\frac{\ln(1.6/1.2)}{2\pi(380\text{ W/m.°C})(1\text{ m})}+\frac{(0.0002\text{ m}^2.°\text{C/W})}{[\pi(0.016\text{ m})(1\text{ m})]}$$
$$+\frac{1}{(1400\text{ W/m}^2.°\text{C})[\pi(0.016\text{ m})(1\text{ m})]}$$
$$=\mathbf{0.06947°C/W}$$

(*b*) The overall heat transfer coefficient based on the inner and the outer surface areas of the tube per length are

$$R=\frac{1}{UA}=\frac{1}{U_i A_i}=\frac{1}{U_o A_o}$$

$$U_i=\frac{1}{RA_i}=\frac{1}{(0.06947\text{ °C/W})[\pi(0.012\text{ m})(1\text{ m})]}=\mathbf{382\ W/m^2.°C}$$

$$U_o=\frac{1}{RA_o}=\frac{1}{(0.06947\text{ °C/W})[\pi(0.016\text{ m})(1\text{ m})]}=\mathbf{286\ W/m^2.°C}$$

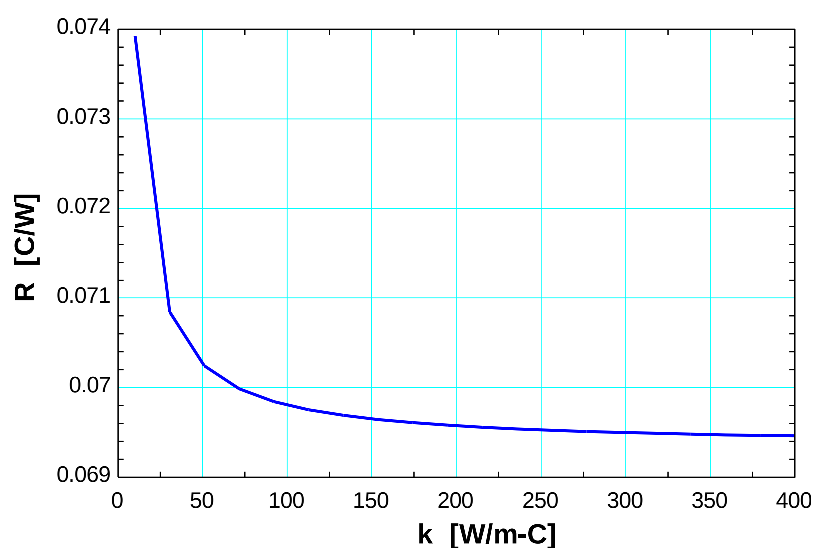
11-27  Prob. 11-26 is reconsidered. The effects of pipe conductivity and heat transfer coefficients on the thermal resistance of the heat exchanger are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

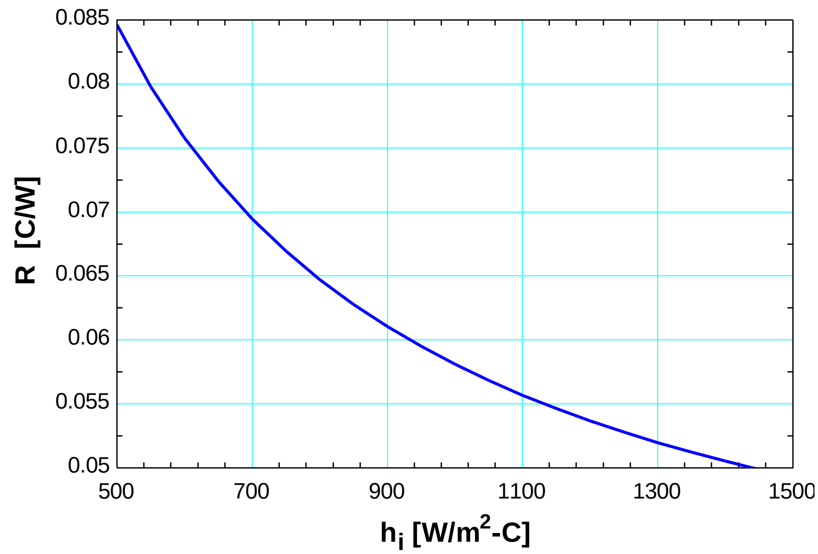
```
"GIVEN"
k=380 [W/m-C]
D_i=0.012 [m]
D_o=0.016 [m]
D_2=0.03 [m]
h_i=700 [W/m^2-C]
h_o=1400 [W/m^2-C]
R_f_i=0.0005 [m^2-C/W]
R_f_o=0.0002 [m^2-C/W]

"ANALYSIS"
R=1/(h_i*A_i)+R_f_i/A_i+ln(D_o/D_i)/(2*pi*k*L)+R_f_o/A_o+1/(h_o*A_o)
L=1 [m] "a unit length of the heat exchanger is considered"
A_i=pi*D_i*L
A_o=pi*D_o*L
U_i=1/(R*A_i)
U_o=1/(R*A_o)
```

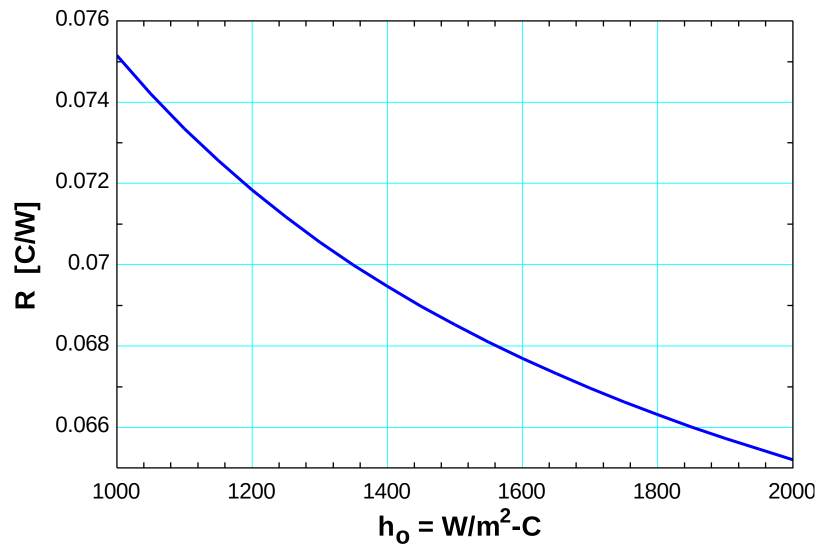
k [W/m-C]	R [C/W]
10	0.07392
30.53	0.07085
51.05	0.07024
71.58	0.06999
92.11	0.06984
112.6	0.06975
133.2	0.06969
153.7	0.06964
174.2	0.06961
194.7	0.06958
215.3	0.06956
235.8	0.06954
256.3	0.06952
276.8	0.06951
297.4	0.0695
317.9	0.06949
338.4	0.06948
358.9	0.06947
379.5	0.06947
400	0.06946



h_i [W/m^2-C]	R [C/W]
500	0.08462
550	0.0798
600	0.07578
650	0.07238
700	0.06947
750	0.06694
800	0.06473
850	0.06278
900	0.06105
950	0.05949
1000	0.0581
1050	0.05684
1100	0.05569
1150	0.05464
1200	0.05368
1250	0.05279
1300	0.05198
1350	0.05122
1400	0.05052
1450	0.04987
1500	0.04926



h_o [W/m^2-C]	R [C/W]
1000	0.07515
1050	0.0742
1100	0.07334
1150	0.07256
1200	0.07183
1250	0.07117
1300	0.07056
1350	0.06999
1400	0.06947
1450	0.06898
1500	0.06852
1550	0.06809
1600	0.06769
1650	0.06731
1700	0.06696
1750	0.06662
1800	0.06631
1850	0.06601
1900	0.06573
1950	0.06546
2000	0.0652



11-28 Hot oil entering a double pipe heat exchanger is cooled by cold water at 20°C. For the known values of oil and water flow rates and fouling factors, the overall heat transfer coefficient on inner and outer surface of the copper tube are to be determined.

Assumptions **1**Steady state conditions exist. **2** Heat exchanger is well insulated. **3** The flow of oil and water is both hydrodynamically and thermally fully developed. **4** Properties of oil and water are constant.

Properties The properties of oil are to be evaluated at the average inlet and exit temperatures of 150°C and 50°C or 100 °C. The properties of oil at an average temperature of 100°C are (Table A-13)

$$\rho = 840\ \text{kg/m}^3,\ c_p = 2220\ \text{J/kg}\cdot\text{K},\ k = 0.1367\ \text{W/m}\cdot\text{K},\ \mu = 0.01718\ \text{kg/m}\cdot\text{s and}\ \ \text{Pr} = 279.1$$

The properties of water evaluated at an average temperature at the average inlet and exit temperatures of 20°C and 70°C or 45°C are (Table A-9)

$$\rho = 990.1\ \text{kg/m}^3,\ c_p = 4180\ \text{J/kg}\cdot\text{K},\ k = 0.637\ \text{W/m}\cdot\text{K},\ \mu = 0.596\times10^{-3}\ \text{kg/m}\cdot\text{s and}\ \ \text{Pr} = 3.91$$

Analysis The overall heat transfer coefficient is determined as,

$$\frac{1}{UA_s} = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o}$$

The internal and outer surface areas of the heat exchanger tube are,

$$A_i = \pi D_i L = \pi(0.02\ \text{m})(1.5\ \text{m}) = 0.0942\ \text{m}^2$$

$$A_o = \pi D_o L = \pi(0.0225\ \text{m})(1.5\ \text{m}) = 0.106\ \text{m}^2$$

We first need to determine the convection heat transfer coefficients at inner and outer pipe surfaces. Based on the mass flow rate and density of oil the average velocity of the oil and the Reynolds number is calculated as follows,

$$V = \frac{4\dot{m}_h}{\rho\pi D_i^2} = \frac{4\,(2\ \text{kg/s})}{(840\ \text{kg/m}^3)\pi\,(0.02\ \text{m})^2} = 7.58\ \text{m/s}$$

The Reynolds number for the flow of oil is,

$$\text{Re} = \frac{\rho V D_i}{\mu} = \frac{(840\ \text{kg/m}^3)(7.58\ \text{m/s})(0.02\text{m})}{0.01718\ \text{kg/m}\cdot\text{s}} = 7412$$

The mass flow rate of cooling water can be determined from the heat balance such that the heat rejected by the hot engine oil is equal to the heat absorbed by the cooling water

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$(444\ \text{W/K})\,(150 - 50)°C = \dot{m}_c\,(4180\text{kJ/kg}\cdot\text{K})\,(70 - 20)°\text{C}$$
$$\dot{m}_c = 2.124\ \text{kg/s}$$

Thus the mass flow rate of cooling water is 2.124 kg/s. The hydraulic diameter of the annular space on the shell side is calculated as,

$$D_h = (D_o - D_i) = (0.06 - 0.0225)\,m = 0.0375\ \text{m}$$

The average velocity of the cooling water is

$$V = \frac{4\dot{m}}{\rho\pi D_h^2} = \frac{4(2.124\ \text{kg/s})}{(990.1\ \text{kg/m}^3)\,\pi(0.0375\text{m})^2} = 1.942\ \text{m/s}$$

The Reynolds number for the flow of water is,

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(990.1\text{kg/m}^3)(1.942\ \text{m/s})(0.0375\text{m})}{0.596\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 120{,}980$$

For both oil and water since $3\times10^3 < \mathrm{Re} < 5\times10^6$ and $0.5 \le \mathrm{Pr} \le 2000$ we can use from Chapter 9, the Gnielinski (1976) correlation to find the Nusselt number and hence the convective heat transfer coefficient

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\sqrt{(f/8)}\left(\mathrm{Pr}^{2/3}-1\right)}$$

where the friction factor f may be calculated from the explicit *first Pteukhov equation* [Pteukhov (1970)] as,

$$f = (0.79\ln\mathrm{Re}-1.64)^{-2}$$

For oil, $f = (0.79\ln(7412) - 1.64)^{-2} = 0.03429$

For water, $f = (0.79\ln(120980) - 1.64)^{-2} = 0.01728$

The Nusselt number for oil and water side are calculate as

For oil,

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\sqrt{(f/8)}\left(\mathrm{Pr}^{2/3}-1\right)} = \frac{0.00428(6412)\times279.1}{1+12.7\times0.0654(41.7)} = 214.94$$

and

$$h = \frac{k}{D}Nu = \frac{0.1367\,\mathrm{W/m\cdot K}}{0.02\,\mathrm{m}}\ (214.94) = 1469.11\,\mathrm{W/m^2\cdot K}$$

For water,

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\sqrt{(f/8)}\left(\mathrm{Pr}^{2/3}-1\right)} = \frac{0.00216(119980)\times3.91}{1+12.7\times0.0464(1.482)} = 540.91$$

and

$$h = \frac{k}{D_h}Nu = \frac{0.637\,\mathrm{W/m\cdot K}}{0.0375\,\mathrm{m}}\ (540.91) = 9188.25\,\mathrm{W/m^2\cdot K}$$

Thus the overall heat transfer coefficient is determined as,

$$\frac{1}{UA_s} = \frac{1}{(1469.1\ \mathrm{W/m^2\cdot K})(0.0942\,\mathrm{m^2})} + \frac{0.00015\,\mathrm{m^2\cdot K/W}}{0.0942\ \mathrm{m^2}} + \frac{\ln(0.0225/0.02)}{2\pi\pi(25\,\mathrm{W/m\cdot K})\,(1.5\,\mathrm{m})} +$$
$$\frac{0.0001\mathrm{m^2\cdot K/W}}{0.106\,\mathrm{m^2}} + \frac{1}{(9188.25\,\mathrm{W/m^2\cdot K})(0.106\,\mathrm{m^2})} = 0.01083\,\mathrm{K/W}$$

Using the relationship

$$\frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o}$$

Thus the overall heat transfer coefficient based on inner surface of the tube is

U_i = **979.4 W/m²·K.**

The overall heat transfer coefficient based on outer surface of the tube is

U_o = **871.09 W/m²·K.**

Analysis of Heat Exchangers

11-29C The heat exchangers usually operate for long periods of time with no change in their operating conditions, and then they can be modeled as steady-flow devices. As such , the mass flow rate of each fluid remains constant and the fluid properties such as temperature and velocity at any inlet and outlet remain constant. The kinetic and potential energy changes are negligible. The specific heat of a fluid can be treated as constant in a specified temperature range. Axial heat conduction along the tube is negligible. Finally, the outer surface of the heat exchanger is assumed to be perfectly insulated so that there is no heat loss to the surrounding medium and any heat transfer thus occurs is between the two fluids only.

11-30C That relation is valid under steady operating conditions, constant specific heats, and negligible heat loss from the heat exchanger.

11-31C The mass flow rate of the cooling water can be determined from $\dot{Q} = (\dot{m}c_p\Delta T)_{\text{coolingwater}}$. The rate of condensation of the steam is determined from $\dot{Q} = (\dot{m}h_{fg})_{\text{steam}}$, and the total thermal resistance of the condenser is determined from $R = \dot{Q}/\Delta T$.

11-32C The product of the mass flow rate and the specific heat of a fluid is called the heat capacity rate and is expressed as $C = \dot{m}c_p$. When the heat capacity rates of the cold and hot fluids are equal, the temperature change is the same for the two fluids in a heat exchanger. That is, the temperature rise of the cold fluid is equal to the temperature drop of the hot fluid. A heat capacity of infinity for a fluid in a heat exchanger is experienced during a phase-change process in a condenser or boiler.

11-33C When the heat capacity rates of the cold and hot fluids are identical, the temperature rise of the cold fluid will be equal to the temperature drop of the hot fluid.

11-34 Hot and cold fluid streams have same heat capacity rates. It is to be proved that the temperature profiles of the hot and cold fluids are parallel to each other at any given section of the heat exchanger.

Assumption **1** Heat exchanger is well insulated. **2** Fluid properties do not change with heat exchanger length.

Analysis Assuming heat exchanger to be well insulated and the heat transfer occurs only between the hot and cold fluid, the heat transfer across the differential section of the heat exchanger can be expressed as,

$$\delta\dot{Q} = -\dot{m}_h c_{ph} dT_h = \dot{m}_c c_{pc} dT_c$$

Thus the rate of heat loss from the hot fluid at any section of the heat exchanger is equal to the rate of heat gain by the cold fluid in that section.

However, in a counter flow heat exchanger, the temperature of both hot and cold stream decreases in the direction of heat exchanger length. Thus we have,

$$\delta\dot{Q} = -\dot{m}_h c_{ph} dT_h = -\dot{m}_c c_{pc} dT_c$$

The above energy balance can be written as,

$$dT_h = -\frac{\delta\dot{Q}}{\dot{m}_h c_{ph}} \text{ and } dT_c = -\frac{\delta\dot{Q}}{\dot{m}_c c_{pc}}$$

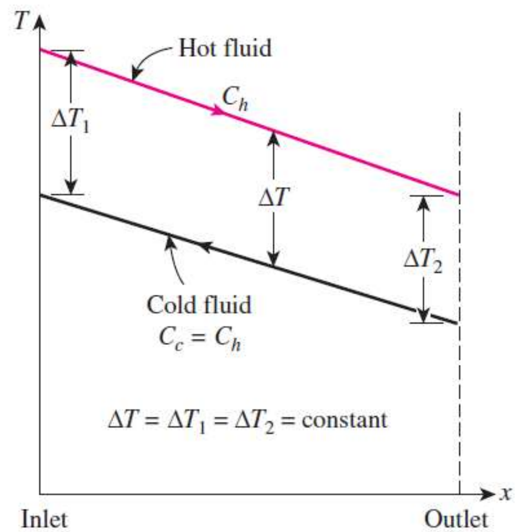
Thus we get,

$$dT_h - dT_c = -\frac{\delta\dot{Q}}{\dot{m}_h c_{ph}} + \frac{\delta\dot{Q}}{\dot{m}_c c_{pc}} = -\delta\dot{Q}\left(\frac{1}{\dot{m}_h c_{ph}} - \frac{1}{\dot{m}_h c_{ph}}\right) = d\Delta T$$

For same heat capacity rates of both hot and cold fluid streams, we have,

$$dT_h - dT_c = d\Delta T = 0$$

This implies that the temperature difference between the hot and cold fluid stream at any given section remains constant. Hence, the temperature profile of two fluid streams (hot and cold) that have same heat capacity rates is parallel to each other at every section of the counter flow heat exchanger as shown in the figure.



The Log Mean Temperature Difference Method

11-35C ΔT_{lm} is called the log mean temperature difference, and is expressed as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

where

$$\Delta T_1 = T_{h,in} - T_{c,in} \qquad \Delta T_2 = T_{h,out} - T_{c,out} \qquad \text{for parallel-flow heat exchangers}$$

and

$$\Delta T \ = T_{h,in} - T_{c,out} \qquad \Delta T_2 = T_{h,out} - T_{c,in} \qquad \text{for counter-flow heat exchangers}$$

11-36C The temperature difference between the two fluids decreases from ΔT_1 at the inlet to ΔT_2 at the outlet, and arithmetic mean temperature difference is defined as $\Delta T_{\mathrm{am}} = \dfrac{\Delta T_1 + \Delta T_2}{2}$. The logarithmic mean temperature difference ΔT_{lm} is obtained by tracing the actual temperature profile of the fluids along the heat exchanger, and is an exact representation of the average temperature difference between the hot and cold fluids. It truly reflects the exponential decay of the local temperature difference. The logarithmic mean temperature difference is always less than the arithmetic mean temperature.

11-37C ΔT_{lm} cannot be greater than both ΔT_1 and ΔT_2 because ΔT_{ln} is always less than or equal to ΔT_{m} (arithmetic mean) which can not be greater than both ΔT_1 and ΔT_2.

11-38C In the parallel-flow heat exchangers the hot and cold fluids enter the heat exchanger at the same end, and the temperature of the hot fluid decreases and the temperature of the cold fluid increases along the heat exchanger. But the temperature of the cold fluid can never exceed that of the hot fluid. In case of the counter-flow heat exchangers the hot and cold fluids enter the heat exchanger from the opposite ends and the outlet temperature of the cold fluid may exceed the outlet temperature of the hot fluid.

11-39C First heat transfer rate is determined from $\dot{Q} = \dot{m}c_p[T_{in} - T_{out}]$, ΔT_{ln} from $\Delta T_{lm} = \dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$, correction factor from the figures, and finally the surface area of the heat exchanger from $\dot{Q} = UAFDT_{lm,CF}$

11-40C The factor *F* is called as correction factor which depends on the geometry of the heat exchanger and the inlet and the outlet temperatures of the hot and cold fluid streams. It represents how closely a heat exchanger approximates a counter-flow heat exchanger in terms of its logarithmic mean temperature difference. *F* cannot be greater than unity.

11-41C In this case it is not practical to use the LMTD method because it requires tedious iterations. Instead, the effectiveness-NTU method should be used.

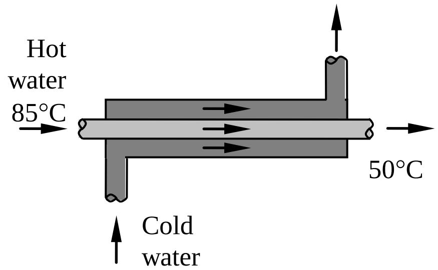
11-42C The $\Delta T_{\rm lm}$ will be greatest for double-pipe counter-flow heat exchangers.

11-43 Water is heated in a double-pipe, parallel-flow uninsulated heat exchanger by geothermal water. The rate of heat transfer to the cold water and the log mean temperature difference for this heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heat of hot water is given to be 4.25 kJ/kg.°C.

Analysis The rate of heat given up by the hot water is



$$\begin{aligned}\dot{Q}_h &= [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \\ &= (1.4\ \text{kg/s})(4.25\ \text{kJ/kg.°C})(85°\text{C} - 50°\text{C}) \\ &= 208.3\ \text{kW}\end{aligned}$$

The rate of heat picked up by the cold water is

$$\dot{Q}_c = (1-0.03)\dot{Q}_h = (1-0.03)(208.3\ \text{kW}) = \mathbf{202.0\ kW}$$

The log mean temperature difference is

$$\dot{Q} = UA\Delta T_{lm} \longrightarrow \Delta T_{lm} = \frac{\dot{Q}}{UA} = \frac{202.0\ \text{kW}}{(1.15\ \text{kW/m}^2\cdot°\text{C})(4\ \text{m}^2)} = \mathbf{43.9°C}$$

11-44 A stream of hydrocarbon is cooled by water in a double-pipe counterflow heat exchanger. The overall heat transfer coefficient is to be determined.

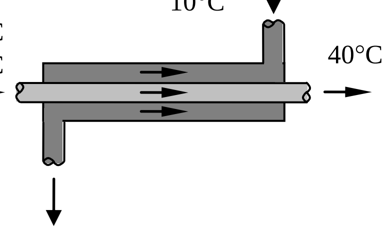
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heats of hydrocarbon and water are given to be 2.2 and 4.18 kJ/kg.°C, respectively.

Analysis The rate of heat transfer is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{HC}} = (720/3600\ \text{kg/s})(2.2\ \text{kJ/kg.°C})(150°\text{C} - 40°\text{C}) = 48.4\ \text{kW}$$

The outlet temperature of water is



$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{w}}$$
$$48.4\ \text{kW} = (540/3600\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(T_{\text{w,out}} - 10°\text{C})$$
$$T_{\text{w,out}} = 87.2\,°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 150°\text{C} - 87.2°\text{C} = 62.8°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 10°\text{C} = 30°\text{C}$$

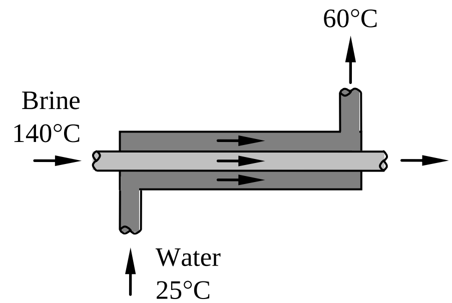
and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{62.8 - 30}{\ln(62.8/30)} = 44.4°\text{C}$$

The overall heat transfer coefficient is determined from

$$\dot{Q} = UA\Delta T_{lm}$$
$$48.4\ \text{kW} = U(\pi \times 0.025 \times 6.0)(44.4°\text{C})$$
$$U = \mathbf{2.31\ kW/m^2 \cdot K}$$

11-45 Water is heated in a double-pipe parallel-flow heat exchanger by geothermal water. The required length of tube is to be determined.



Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in the heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (0.2\text{ kg/s})(4.18\text{ kJ/kg.°C})(60°\text{C} - 25°\text{C}) = 29.26\text{ kW}$$

Then the outlet temperature of the geothermal water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{geot.water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 140°\text{C} - \frac{29.26\text{ kW}}{(0.3\text{ kg/s})(4.31\text{ kJ/kg.°C})} = 117.4°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,in} = 140°\text{C} - 25°\text{C} = 115°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,out} = 117.4°\text{C} - 60°\text{C} = 57.4°\text{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{115 - 57.4}{\ln(115/57.4)} = 82.9°\text{C}$$

The surface area of the heat exchanger is determined from

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{29.26\text{ kW}}{(0.55\text{ kW/m}^2)(82.9°\text{C})} = 0.642\text{ m}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.642\text{ m}^2}{\pi(0.008\text{ m})} = \mathbf{25.5\,m}$$

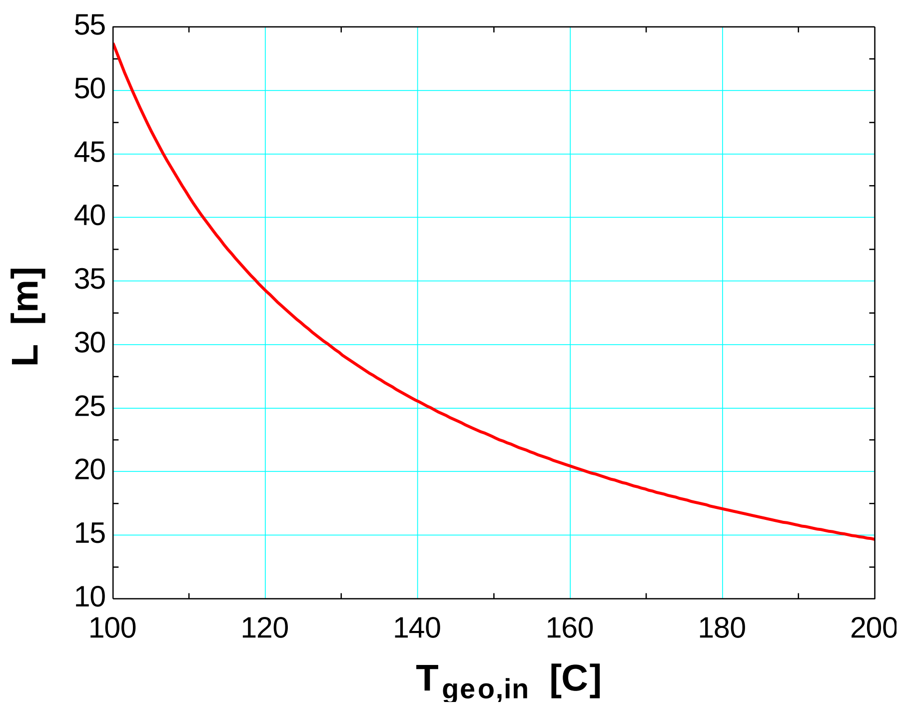
11-46 Prob. 11-45 is reconsidered. The effects of temperature and mass flow rate of geothermal water on the length of the tube are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

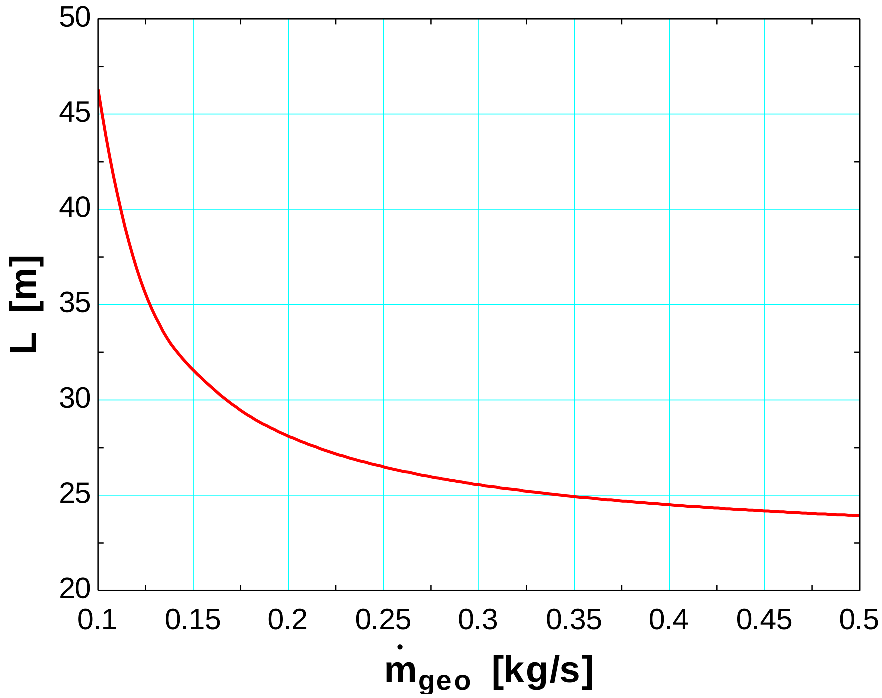
```
"GIVEN"
T_w_in=25 [C]
T_w_out=60 [C]
m_dot_w=0.2 [kg/s]
c_p_w=4.18 [kJ/kg-C]
T_geo_in=140 [C]
m_dot_geo=0.3 [kg/s]
c_p_geo=4.31 [kJ/kg-C]
D=0.008 [m]
U=0.55 [kW/m^2-C]

"ANALYSIS"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
Q_dot=m_dot_geo*c_p_geo*(T_geo_in-T_geo_out)
DELTAT_1=T_geo_in-T_w_in
DELTAT_2=T_geo_out-T_w_out
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
A=pi*D*L
```

$T_{geo,in}$ [C]	L [m]
100	53.73
105	46.81
110	41.62
115	37.56
120	34.27
125	31.54
130	29.24
135	27.26
140	25.54
145	24.04
150	22.7
155	21.51
160	20.45
165	19.48
170	18.61
175	17.81
180	17.08
185	16.4
190	15.78
195	15.21
200	14.67



$\dot{m}_{geo}$ [kg/s]	L [m]
0.1	46.31
0.125	35.52
0.15	31.57
0.175	29.44
0.2	28.1
0.225	27.16
0.25	26.48
0.275	25.96
0.3	25.54
0.325	25.21
0.35	24.93
0.375	24.69
0.4	24.49
0.425	24.32
0.45	24.17
0.475	24.04
0.5	23.92



11-47 Glycerin is heated by ethylene glycol in a thin-walled double-pipe parallel-flow heat exchanger. The rate of heat transfer, the outlet temperature of the glycerin, and the mass flow rate of the ethylene glycol are to be determined.

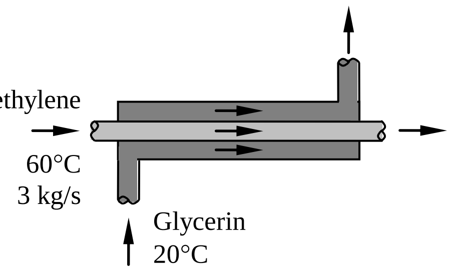
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of glycerin and ethylene glycol are given to be 2.4 and 2.5 kJ/kg.°C, respectively.

Analysis (*a*) The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,in} = 60°\text{C} - 20°\text{C} = 40°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out} = T_{h,out} - (T_{h,out} - 15°\text{C}) = 15°\text{C}$$



and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40-15}{\ln(40/15)} = 25.49°\text{C}$$

Then the rate of heat transfer becomes

$$\dot{Q} = UA_s \Delta T_{lm} = (240\ \text{W/m}^2.°\text{C})(3.2\ \text{m}^2)(25.49°\text{C}) = 19{,}576\ \text{W} = \mathbf{19.58\ kW}$$

(*b*) The outlet temperature of the glycerin is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{glycerin}} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{19.58\ \text{kW}}{(0.5\ \text{kg/s})(2.4\ \text{kJ/kg.°C})} = \mathbf{36.32°C}$$

(*c*) Then the mass flow rate of ethylene glycol becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{ethyleneglycol}}$$

$$\dot{m}_{\text{ethyleneglycol}} = \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{19.58\ \text{kJ/s}}{(2.5\ \text{kJ/kg.°C})[60°\text{C} - (36.32+15)°\text{C}]} = \mathbf{0.902\ kg/s}$$

11-48 The heat transfer rate of a heat exchanger containing 400 tubes with specified inner and outer diameters and length is to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible. **5** Thermal resistance of the tubes is negligible.

Analysis The overall heat transfer coefficient based on the outer surface is

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \quad \rightarrow \quad \frac{1}{U_o} = \frac{A_o}{h_i A_i} + \frac{1}{h_o} = \frac{\pi D_o L}{\pi D_i L h_i} + \frac{1}{h_o} = \frac{D_o}{D_i}\frac{1}{h_i} + \frac{1}{h_o}$$

or

$$U_o = \left[\frac{D_o}{D_i}\frac{1}{h_i} + \frac{1}{h_o}\right]^{-1} = \left[\left(\frac{25}{23}\right)\left(\frac{1}{3410}\right) + \frac{1}{6820}\right]^{-1} \text{ W/m}^2\cdot\text{K} = 2149 \text{ W/m}^2\cdot\text{K}$$

The heat transfer rate is

$$\begin{aligned}\dot{Q} &= U_o A_o \Delta T_{\text{lm}} \\ &= n U_o \pi D_o L \Delta T_{\text{lm}} \\ &= (400)(2149 \text{ W/m}^2\cdot\text{K})\pi(0.025\text{ m})(3.7\text{ m})(23\text{ K}) \\ &= \mathbf{5.75\times 10^6\ W}\end{aligned}$$

Discussion If the inner to outer diameter ratio is neglected, the overall heat transfer coefficient based on the outer surface area becomes

$$U_o = \left[\frac{1}{h_i} + \frac{1}{h_o}\right]^{-1} = 2273 \text{ W/m}^2\cdot\text{K}$$

which is about 6% larger than the original value of $U_o = 2149 \text{ W/m}^2\cdot\text{K}$.

11-49E The required number of tubes and length of tubes for a single pass heat exchanger to heat 100,000 lbm of water in an hour from 60°F to 100°F are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible. **5** Thermal resistance of the tubes is negligible.

Properties The density and specific heat of water are given to be 62.3 lbm/ft^3 and c_{pc} = 1 Btu/lbm·°F, respectively.

Analysis From the equation for mass flow rate, we have

$$\dot{m}_c = n\rho AV \quad \rightarrow \quad n = \frac{\dot{m}_c}{\rho AV}$$

$$n = \frac{\dot{m}_c}{\rho AV} = \frac{100{,}000\ \text{lbm/hr}}{(62.3\ \text{lbm/ft}^2)(\pi/4)(1.2/12\ \text{ft})^2(4\ \text{ft/s})(3600\ \text{s/hr})} = 14.19$$

Hence, the number of tubes required to heat 100,000 lbm of water in an hour is

$$n = \mathbf{15\ tubes}$$

The overall heat transfer coefficient based on the inner surface is

$$\frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o} \quad \rightarrow \quad \frac{1}{U_i} = \frac{A_i}{h_i A_i} + \frac{A_i}{h_o A_o} = \frac{1}{h_i} + \frac{D_i}{D_o}\frac{1}{h_o}$$

where

$$D_o = D_i + 2t = 1.44\ \text{in.}$$

Hence,

$$U_i = \left[\frac{1}{h_i} + \frac{D_i}{D_o}\frac{1}{h_o}\right]^{-1} = \left[\frac{1}{480} + \left(\frac{1.2}{1.44}\right)\frac{1}{2000}\right]^{-1} \text{Bth/hr}\cdot\text{ft}^2\cdot°\text{F} = 400\ \text{Bth/hr}\cdot\text{ft}^2\cdot°\text{F}$$

The log mean temperature difference is

$$\Delta T_{\text{lm,}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(230-60)-(230-100)}{\ln[(230-60)/(230-100)]}\,°\text{F} = 149.1°\text{F}$$

Using the equation for the heat transfer rate, we have

$$\dot{Q} = U_i A_i \Delta T_{\text{lm}} = nU_i \pi D_i L \Delta T_{\text{lm}} \quad \rightarrow \quad L = \frac{\dot{Q}}{nU_i \pi D_i \Delta T_{\text{lm}}} = \frac{\dot{m}_c c_{pc}(T_{c,\text{out}} - T_{c,\text{in}})}{nU_i \pi D_i \Delta T_{\text{lm}}}$$

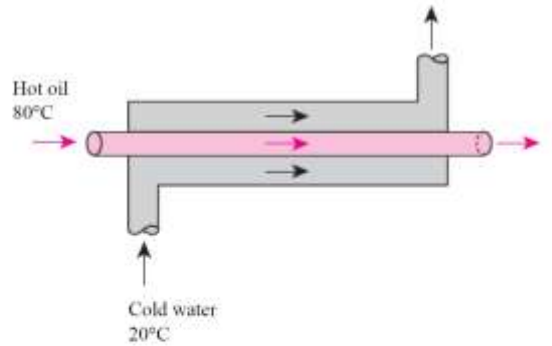
$$L = \frac{(100{,}000\ \text{lbm/hr})(1\ \text{Btu/lbm}\cdot°\text{F})(100-60)°\text{F}}{(15)(400\ \text{Bth/hr}\cdot\text{ft}^2\cdot°\text{F})\pi(1.2/12\ \text{ft})(149.1°\text{F})} = \mathbf{14.2\ ft}$$

Discussion For process involving condensation, we have $T_{h,\text{in}} = T_{h,\text{out}}$.

11-50 Hot engine oil and cooling water flows through a parallel heat exchanger at specified temperatures. The mass flow rate of the cooling water, log mean temperature difference and the area of heat exchanger are to be determined.

Assumption **1** Thermal resistance of the thin walled copper tube is negligible. **2** Thermal properties of oil and water are constant. **3** Heat exchanger is well insulated.

Analysis The heat transfer rate though the heat exchanger is



$$\begin{aligned}\dot{Q} &= \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) \\ &= (0.3\,\text{kg/s})(2048\,\text{J/kg}\cdot\text{K})(80-40)^{\text{o}}\text{C} \\ &= 24.576\ \text{kW}\end{aligned}$$

(a) From energy balance,

Rate of heat loss by engine oil = Rate of heat gain by cooling water

Thus we have,

$$\dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\dot{m}_c = \dot{m}_h \frac{c_{ph}(T_{h,in} - T_{h,out})}{c_{pc}(T_{c,out} - T_{c,in})} = (0.3\,\text{kg/s})\frac{(2048\,\text{J/kg}\cdot\text{K})(80-40)^{\text{o}}\text{C}}{(4180\text{J/kg}\cdot\text{K})(32-20)^{\text{o}}\text{C}} = \mathbf{0.489\ kg/s}$$

(b) The log mean temperature difference is calculated based on the inlet and exit temperatures of the hot and cold fluids. The temperature difference between the two fluids at inlet and exit of the heat exchanger are calculated as follows.

$$\Delta T_1 = T_{h,in} - T_{c,in} = 80 - 20 = 60\ {}^{o}C$$

and

$$\Delta T_2 = T_{h,out} - T_{c,out} = 40 - 32 = 8^{o}C$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{60-8}{\ln(60/8)} = \mathbf{25.8\ {}^{o}C}$$

(c) The heat transfer rate in the heat exchanger is also calculated as,

$$\dot{Q} = UA_s\Delta T_{lm}$$

The overall heat transfer coefficient is calculated as,

$$U = \left(\frac{1}{h_i} + \frac{1}{h_o}\right)^{-1} = \left(\frac{1}{750} + \frac{1}{350}\right)^{-1} = 238.6\,\text{W/m}^2\cdot\text{K}$$

Thus the surface area of the heat exchanger is,

$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{24576\,\text{W}}{(238.6\,\text{W/m}^2\cdot\text{K})(25.8^{\text{o}}\text{C})} = \mathbf{3.99\ m^2}$$

11-51 The hot and cold fluid streams at specified temperature and mass flow rate enter a parallel heat exchanger. For the known values of convection heat transfer coefficients and fouling factors, the overall heat transfer coefficient, the exit temperature of the hot fluid and the surface area of the heat exchanger are to be determined.

Assumptions **1** Thermal resistance due to pipe thickness is negligible. **2** Thermal properties of the hot and cold fluids are constant. **3** Heat exchanger is well insulated.

Analysis (a) The overall heat transfer coefficient is calculated based on the given heat transfer coefficients and fouling factors at the tube inlet and outlet surface. Since the tube is of negligible thickness, the inner and outer surface areas of the tube may be assumed to be equal.

$$\frac{1}{U}=\frac{1}{h_i}+R_{f,i}+R_{f,o}+\frac{1}{h_o}=\frac{1}{300\ \text{W/m}^2\cdot\text{K}}+(0.0003\ \text{m}^2\cdot\text{K/W})+(0.0001\ \text{m}^2\cdot\text{K/W})+\frac{1}{800\ \text{W/m}^2\cdot\text{K}}$$

$$U=\mathbf{200.67\ \ W/m^2\cdot K}$$

(b) Rate of heat loss from hot fluid is

$$\dot{Q}=\dot{m}_h c_{ph}(T_{h,in}-T_{h,out})=(3\,\text{kg/s})(1150\ \text{J/kg}\cdot\text{K})\,(150^{\text{o}}\,\text{C}-T_{h,out})$$

Rate of heat gain by the cold fluid is,

$$\dot{Q}=\dot{m}_c c_{pc}(T_{c,out}-T_{c,in})=(0.5\,\text{kg/s})\,(4180\ \text{J/kg}\cdot\text{K})\,(70-30)^{\text{o}}\,\text{C}=83.6\ \text{kW}$$

From energy balance we have

$$83.6\,\text{kW}=(3450\ \text{W/K})(150^{\text{o}}\,\text{C}-T_{h,out})$$

Thus we get the exit temperature of the hot side fluid to be

$$T_{h,out}=150-\frac{83.6\times10^3\ \text{W}}{3450\ \text{W/K}}=\mathbf{125.8\ ^oC}$$

(c) Using the concept of log mean temperature difference, the heat transfer rate from the heat exchanger can also be calculated as

$$\dot{Q}=UA_s\Delta T_{lm}$$

where

$$\Delta T_{lm}=\frac{\Delta T_1-\Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1=T_{h,in}-T_{c,in}=150-30=120\ ^{\text{o}}\text{C}$$

and

$$\Delta T_2=T_{h,out}-T_{c,out}=125.8-70=55.8\ ^{\text{o}}\text{C}$$

Thus the equation for log mean temperature difference is written as

$$\Delta T_{lm}=\frac{120-55.8}{\ln(120/55.8)}=83.84\ ^{\text{o}}\text{C}$$

Thus the surface area of the heat exchanger is,

$$A_s=\frac{\dot{Q}}{U\Delta T_{lm}}=\frac{83.6\times10^3\ \text{W}}{(200.67\ \text{W/m}^2\cdot\text{K})\,(83.84\ ^{\text{o}}\text{C})}=\mathbf{4.97\,m^2}$$

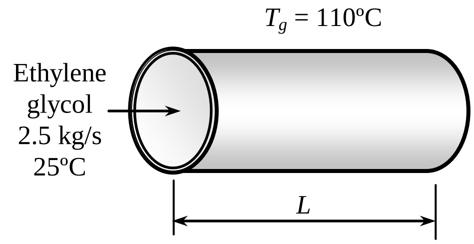
11-52 Ethylene glycol is heated in a tube while steam condenses on the outside tube surface. The tube length is to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the tubes are smooth. **3** Heat transfer to the surroundings is negligible.

Properties The properties of ethylene glycol are given to be ρ = 1109 kg/m^3, c_p = 2428 J/kg·K, k = 0.253 W/m·K, μ = 0.01545 kg/m·s, Pr = 148.5. The thermal conductivity of copper is given to be 386 W/m·K.

Analysis The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (2.5\text{ kg/s})(2428\text{ J/kg.°C})(40-25)\text{°C} = 91{,}050\text{ W}$$



The fluid velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{2.5\text{ kg/s}}{(1109\text{ kg/m}^3)\left[\pi(0.02\text{ m})^2/4\right]} = 7.176\text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(1109\text{ kg/m}^3)(7.176\text{ m/s})(0.02\text{ m})}{0.01545\text{ kg/m}\cdot\text{s}} = 10{,}302$$

which is greater than 10,000. Therefore, we have fully developed flow and evaluate the Nusselt number from turbulent flow relation:

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}302)^{0.8}(148.5)^{0.4} = 275.9$$

Heat transfer coefficient on the inner surface is

$$h_i = \frac{k}{D}Nu = \frac{0.253\text{ W/m.°C}}{0.02\text{ m}}(275.9) = 3490\text{ W/m}^2\text{.°C}$$

Assuming a wall temperature of 100°C, the heat transfer coefficient on the outer surface is determined to be

$$h_o = 9200(T_g - T_w)^{-0.25} = 9200(110-100)^{-0.25} = 5174\text{ W/m}^2\text{.°C}$$

Let us check if the assumption for the wall temperature holds:

$$h_i A_i(T_w - T_{b,\text{avg}}) = h_o A_o(T_g - T_w)$$

$$h_i \pi D_i L(T_w - T_{b,\text{avg}}) = h_o \pi D_o L(T_g - T_w)$$

$$3490\times 0.02(T_w - 32.5) = 5174\times 0.025(110 - T_w) \longrightarrow T_w = 82.84\text{°C}$$

Now we assume a wall temperature of 80°C:

$$h_o = 9200(T_g - T_w)^{-0.25} = 9200(110-80)^{-0.25} = 3931\text{ W/m}^2\text{.°C}$$

Again checking, $3490\times 0.02(T_w - 30) = 3931\times 0.025(110 - T_w) \longrightarrow T_w = 77.8\text{°C}$

which is sufficiently close to the assumed value of 80°C. Now that both heat transfer coefficients are available, we use thermal resistance concept to find overall heat transfer coefficient based on the outer surface area as follows:

$$U_o = \frac{1}{\dfrac{D_o}{h_i D_i} + \dfrac{D_o \ln(D_2/D_1)}{2k_{\text{copper}}} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{0.025}{(3490)(0.02)} + \dfrac{(0.025)\ln(2.5/2)}{2(386)} + \dfrac{1}{3931}} = 1613\text{ W/m}^2\cdot\text{C}$$

The rate of heat transfer can be expressed as

$$\dot{Q} = U_o A_o \Delta T_{\ln}$$

where the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{(T_g - T_e) - (T_g - T_i)}{\ln\left(\dfrac{T_g - T_e}{T_g - T_i}\right)} = \frac{(110-40)-(110-25)}{\ln\left(\dfrac{110-40}{110-25}\right)} = 77.26\text{°C}$$

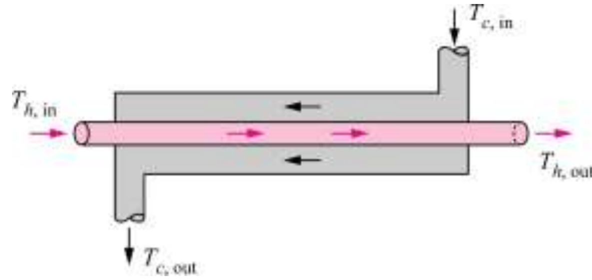
Substituting, the tube length is determined to be

$$\dot{Q} = U_o A_o \Delta T_{lm} \longrightarrow 91{,}050 = (1613)\pi(0.025)L(77.26) \longrightarrow L = \mathbf{9.30\,m}$$

11-53 A counter-flow heat exchanger has a specified overall heat transfer coefficient operating at design and clean conditions. After a period of use built-up scale gives a fouling factor, (*a*) the rate of heat transfer in the heat exchanger and (*b*) the mass flow rates of both hot and cold fluids are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat transfer coefficients and the fouling factors are constant and uniform. **3** Fluid properties are constant.

Properties The specific heat of both hot and cold fluids is given as 4.2 kJ/kg·K.



Analysis When operating at design and clean conditions, the overall heat transfer coefficient is given as

$$U_{\text{w/o scale}} = 284\ \text{W/m}^2\cdot\text{K}$$

(*a*) After a period of use, the overall heat transfer coefficient due to the scale built-up is

$$\frac{1}{U_{\text{w/ scale}}} = \frac{1}{U_{\text{w/o scale}}} + R_f = \frac{1}{284\ \text{W/m}^2\cdot\text{K}} + 0.0004\ \text{m}^2\cdot\text{K/W} = 0.00392\ \text{m}^2\cdot\text{K/W}$$

or

$$U_{\text{w/ scale}} = 255\ \text{W/m}^2\cdot\text{K}$$

The log mean temperature difference is

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(93-38)-(71-27)}{\ln[(93-38)/(71-27)]}\ ^\circ\text{C} = 49.3^\circ\text{C}$$

Then, the rate of heat transfer in the heat exchanger is

$$\dot{Q} = UA_s\Delta T_{\text{lm}} = (255\ \text{W/m}^2\cdot\text{K})(93\ \text{m}^2)(49.3\ \text{K}) = \mathbf{1.17\times10^6\ W}$$

(*b*) The mass flow rate of the hot fluid is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,\text{in}} - T_{h,\text{out}}) \qquad \rightarrow \qquad \dot{m}_h = \frac{\dot{Q}}{c_{ph}(T_{h,\text{in}} - T_{h,\text{out}})}$$

$$\dot{m}_h = \frac{1.17\times10^6\ \text{J/s}}{(4200\ \text{J/kg}\cdot\text{K})(93-71)\ \text{K}} = \mathbf{12.7\ kg/s}$$

The mass flow rate of the cold fluid is

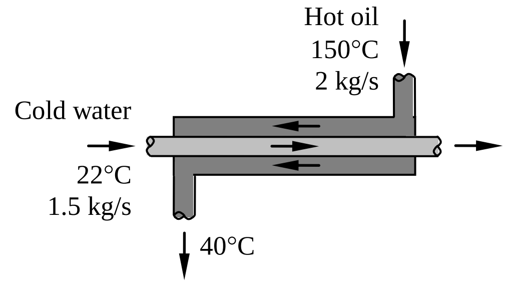
$$\dot{m}_c = \frac{\dot{Q}}{c_{pc}(T_{c,\text{out}} - T_{c,\text{in}})} = \frac{1.17\times10^6\ \text{J/s}}{(4200\ \text{J/kg}\cdot\text{K})(38-27)\ \text{K}} = \mathbf{25.3\ kg/s}$$

Discussion The scale built-up caused a decrease in the overall heat transfer coefficient of the heat exchanger, which reduces the heat removal capability of the heat exchanger.

11-54 Oil is cooled by water in a thin-walled double-pipe counter-flow heat exchanger. The overall heat transfer coefficient of the heat exchanger is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.



Analysis The rate of heat transfer from the water to the oil is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \\ &= (2\text{ kg/s})(2.2\text{ kJ/kg.°C})(150\text{°C} - 40\text{°C}) \\ &= 484\text{ kW}\end{aligned}$$

The outlet temperature of the water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p}$$

$$= 22\text{°C} + \frac{484\text{ kW}}{(1.5\text{ kg/s})(4.18\text{ kJ/kg.°C})} = 99.19\text{°C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 150\text{°C} - 99.19\text{°C} = 50.81\text{°C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40\text{°C} - 22\text{°C} = 18\text{°C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{50.81 - 18}{\ln(50.81/18)} = 31.62\text{°C}$$

Then the overall heat transfer coefficient becomes

$$U = \frac{\dot{Q}}{A_s \Delta T_{lm}} = \frac{484\text{ kW}}{\pi(0.025\text{ m})(6\text{ m})(31.62\text{°C})} = \mathbf{32.5\,kW/m^2.°C}$$

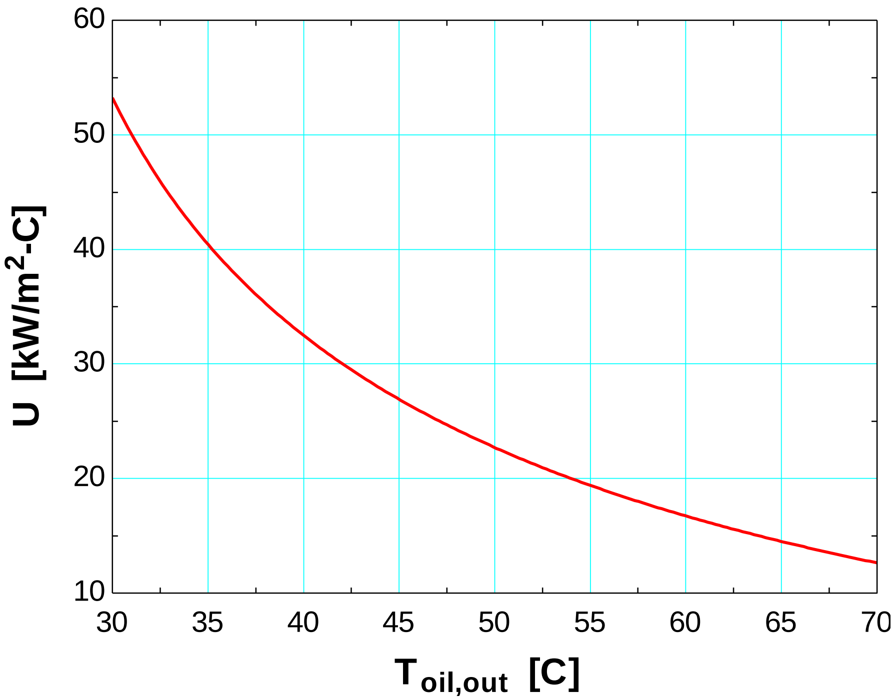
11-55 Prob. 11-54 is reconsidered. The effects of oil exit temperature and water inlet temperature on the overall heat transfer coefficient of the heat exchanger are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

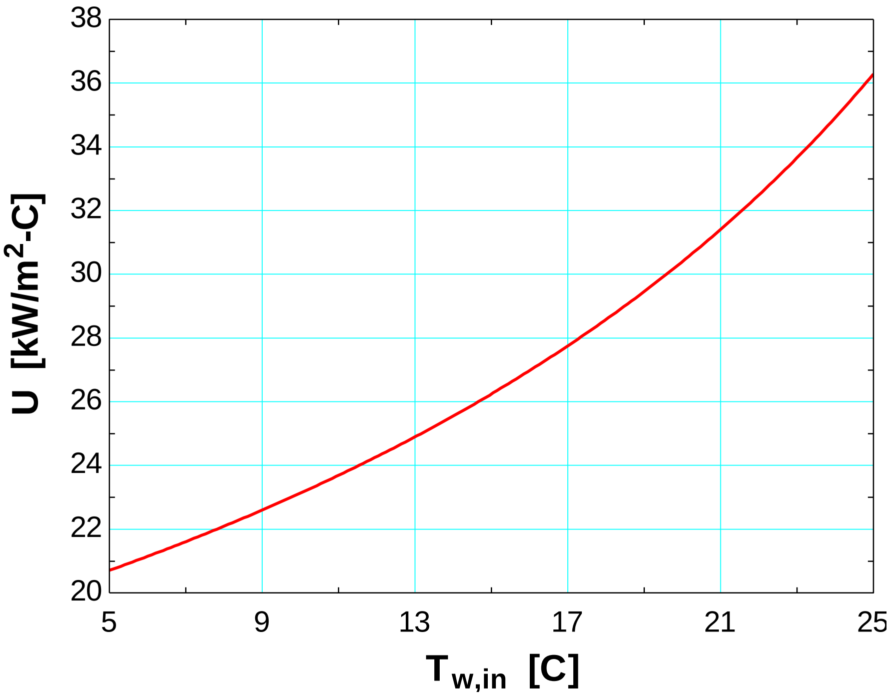
```
"GIVEN"
T_oil_in=150 [C]
T_oil_out=40 [C]
m_dot_oil=2 [kg/s]
c_p_oil=2.20 [kJ/kg-C]
T_w_in=22 [C]
m_dot_w=1.5 [kg/s]
C_p_w=4.18 [kJ/kg-C]
D=0.025 [m]
L=6 [m]

"ANALYSIS"
Q_dot=m_dot_oil*c_p_oil*(T_oil_in-T_oil_out)
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
DELTAT_1=T_oil_in-T_w_out
DELTAT_2=T_oil_out-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
A=pi*D*L
```

$T_{oil,out}$ [C]	U [kW/m^2-C]
30	53.22
32.5	45.94
35	40.43
37.5	36.07
40	32.49
42.5	29.48
45	26.9
47.5	24.67
50	22.7
52.5	20.96
55	19.4
57.5	18
60	16.73
62.5	15.57
65	14.51
67.5	13.53
70	12.63



$T_{w,in}$ [C]	U [kW/m^{2}C]
5	20.7
6	21.15
7	21.61
8	22.09
9	22.6
10	23.13
11	23.69
12	24.28
13	24.9
14	25.55
15	26.24
16	26.97
17	27.75
18	28.58
19	29.46
20	30.4
21	31.4
22	32.49
23	33.65
24	34.92
25	36.29



11-56 Engine oil is heated by condensing steam in a condenser. The rate of heat transfer and the length of the tube required are to be determined.

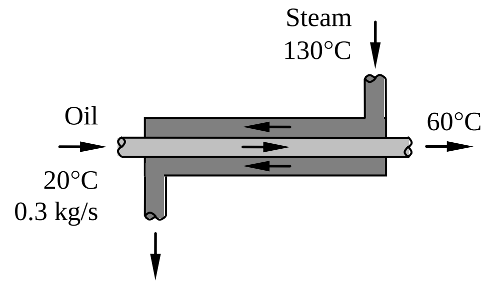
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heat of engine oil is given to be 2.1 kJ/kg.°C. The heat of condensation of steam at 130°C is given to be 2174 kJ/kg.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{oil}} = (0.3\,\text{kg/s})(2.1\,\text{kJ/kg.°C})(60°\text{C} - 20°\text{C}) = \mathbf{25.2\,kW}$$

The temperature differences at the two ends of the heat exchanger are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 130°\text{C} - 60°\text{C} = 70°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 130°\text{C} - 20°\text{C} = 110°\text{C}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{70 - 110}{\ln(70/110)} = 88.5°\text{C}$$

The surface area is

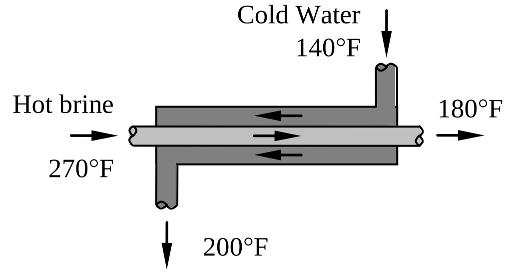
$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{25.2\,\text{kW}}{(0.65\,\text{kW/m}^2.°\text{C})(88.5°\text{C})} = 0.44\,\text{m}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.44\,\text{m}^2}{\pi(0.02\,\text{m})} = \mathbf{7.0\,m}$$

11-57E Water is heated by geothermal water in a double-pipe counter-flow heat exchanger. The mass flow rate of each fluid and the total thermal resistance of the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.



Properties The specific heats of water and geothermal fluid are given to be 1.0 and 1.03 Btu/lbm.°F, respectively.

Analysis The mass flow rate of each fluid is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}}$$

$$\dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{40\ \text{Btu/s}}{(1.0\ \text{Btu/lbm.°F})(200°\text{F} - 140°\text{F})} = \mathbf{0.667\ lbm/s}$$

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{geo.water}}$$

$$\dot{m}_{\text{geo.water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{40\ \text{Btu/s}}{(1.03\ \text{Btu/lbm.°F})(270°\text{F} - 180°\text{F})} = \mathbf{0.431\ lbm/s}$$

The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 270°\text{F} - 200°\text{F} = 70°\text{F}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 180°\text{F} - 140°\text{F} = 40°\text{F}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{70 - 40}{\ln(70/40)} = 53.61°\text{F}$$

Then

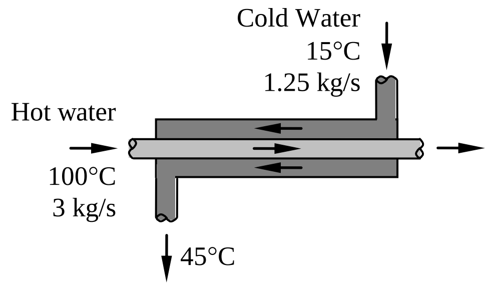
$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow UA_s = \frac{\dot{Q}}{\Delta T_{lm}} = \frac{40\ \text{Btu/s}}{53.61°\text{F}} = 0.7462\ \text{Btu/s.}^{\text{o}}\ \text{F}$$

$$U = \frac{1}{RA_s} \longrightarrow R = \frac{1}{UA_s} = \frac{1}{0.7462\ \text{Btu/s.°F}} = \mathbf{1.34\ s \cdot °F/Btu}$$

11-58 Cold water is heated by hot water in a double-pipe counter-flow heat exchanger. The rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.



Analysis The rate of heat transfer in this heat exchanger is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}} \\ &= (1.25\text{ kg/s})(4.18\text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) \\ &= \mathbf{156.8\ kW}\end{aligned}$$

The outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{156.8\text{ kW}}{(3\text{ kg/s})(4.19\text{ kJ/kg.°C})} = 87.53°\text{C}$$

The temperature differences at the two ends of the heat exchanger are

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out} = 100°\text{C} - 45°\text{C} = 55°\text{C} \\ \Delta T_2 &= T_{h,out} - T_{c,in} = 87.53°\text{C} - 15°\text{C} = 72.53°\text{C}\end{aligned}$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{55 - 72.53}{\ln(55/72.53)} = 63.36°\text{C}$$

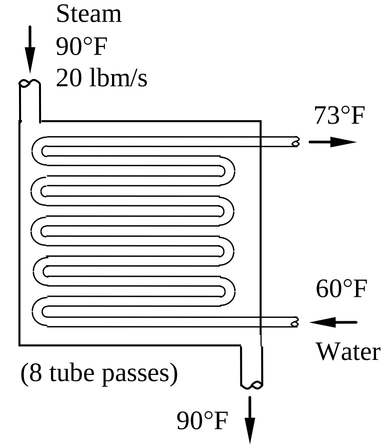
Then the surface area of this heat exchanger becomes

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{156.8\text{ kW}}{(0.880\text{ kW/m}^2.°\text{C})(63.36°\text{C})} = \mathbf{2.81\ m^2}$$

11-59E Steam is condensed by cooling water in a condenser. The rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant. **6** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties We take specific heat of water are given to be 1.0 Btu/lbm.°F. The heat of condensation of steam at 90°F is 1043 Btu/lbm.



Analysis (*a*) The log mean temperature difference is determined from

$$\Delta T_1 = T_{h,in} - T_{c,out} = 90°\text{F} - 73°\text{F} = 17°\text{F}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 90°\text{F} - 60°\text{F} = 30°\text{F}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{17 - 30}{\ln(17/30)} = 22.89°\text{F}$$

The heat transfer surface area is

$$A_s = 8n\pi DL = 8 \times 50 \times \pi(3/48\text{ ft})(5\text{ ft}) = 392.7\text{ ft}^2$$

and

$$\dot{Q} = UA_s\Delta T_{lm} = (600\text{ Btu/h.ft}^2.°\text{F})(392.7\text{ ft}^2)(22.89°\text{F}) = \mathbf{5.393 \times 10^6\ Btu/h} = \mathbf{1498 Btu/s}$$

(*b*) The rate of condensation of the steam is

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{1498\text{ Btu/s}}{1043\text{ Btu/lbm}} = \mathbf{1.44 lbm/s}$$

(*c*) Then the mass flow rate of cold water becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}}$$

$$\dot{m}_{\text{cold water}} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{1498\text{ Btu/s}}{(1.0\text{ Btu/lbm.°F})(73°\text{F} - 60°\text{F}]} = \mathbf{115 lbm/s}$$

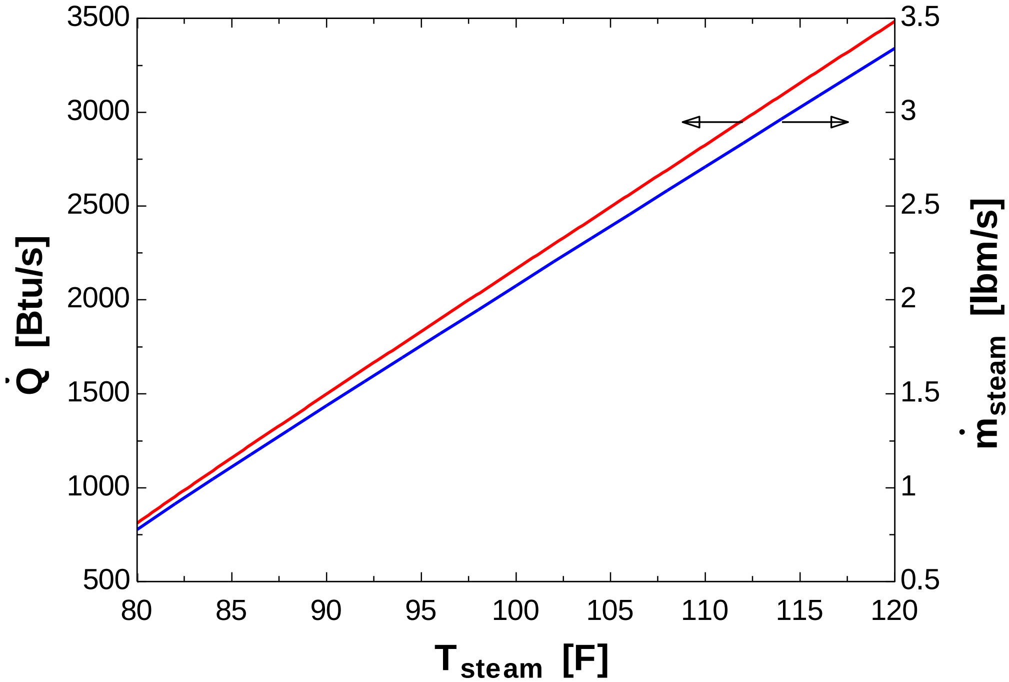
11-60E Prob. 11-59E is reconsidered. The effect of the condensing steam temperature on the rate of heat transfer, the rate of condensation of steam, and the mass flow rate of cold water is to be investigated.

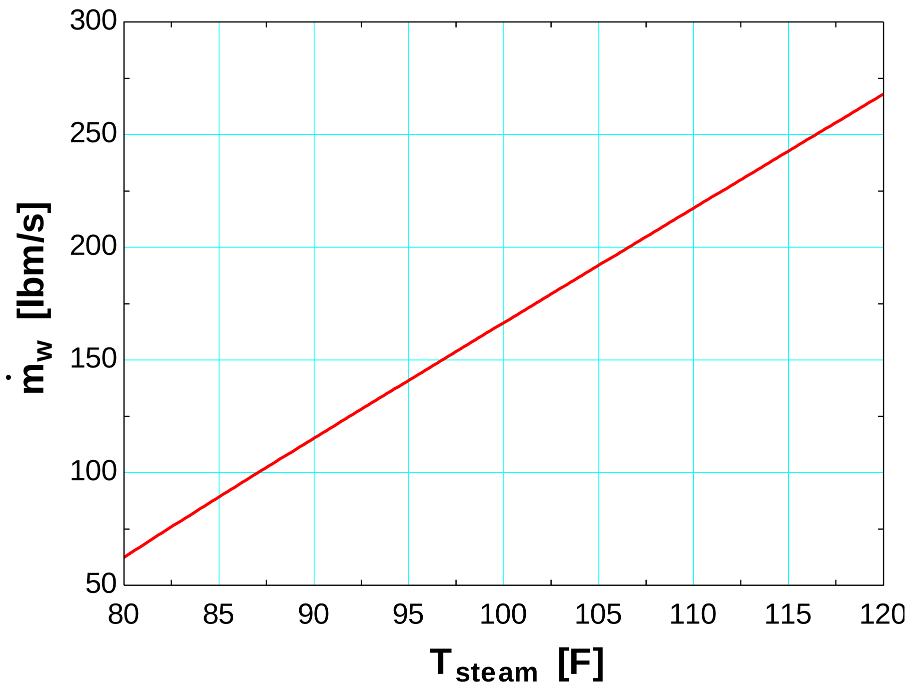
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
N_pass=8
N_tube=50
T_steam=90 [F]
h_fg_steam=1043 [Btu/lbm]
T_w_in=60 [F]
T_w_out=73 [F]
c_p_w=1.0 [Btu/lbm-F]
D=3/4*1/12 [ft]
L=5 [ft]
U=600 [Btu/h-ft^2-F]

"ANALYSIS"
"(a)"
DELTAT_1=T_steam-T_w_out
DELTAT_2=T_steam-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
A=N_pass*N_tube*pi*D*L
Q_dot=U*A*DELTAT_lm*Convert(Btu/h, Btu/s)
"(b)"
Q_dot=m_dot_steam*h_fg_steam
"(c)"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
```

T_{steam} [F]	$\dot{Q}$ [Btu/s]	$\dot{m}_{steam}$ [lbm/s]	$\dot{m}_w$ [lbm/s]
80	810.5	0.7771	62.34
82	951.9	0.9127	73.23
84	1091	1.046	83.89
86	1228	1.177	94.42
88	1363	1.307	104.9
90	1498	1.436	115.2
92	1632	1.565	125.6
94	1766	1.693	135.8
96	1899	1.821	146.1
98	2032	1.948	156.3
100	2165	2.076	166.5
102	2297	2.203	176.7
104	2430	2.329	186.9
106	2562	2.456	197.1
108	2694	2.583	207.2
110	2826	2.709	217.4
112	2958	2.836	227.5
114	3089	2.962	237.6
116	3221	3.088	247.8
118	3353	3.214	257.9
120	3484	3.341	268

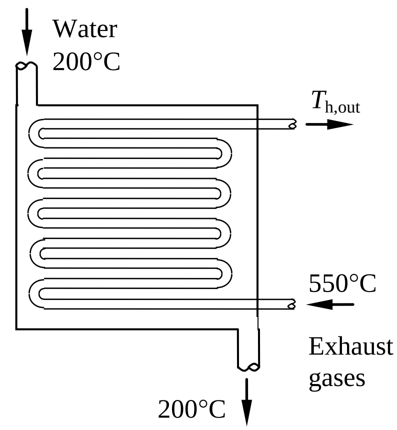

3500
3000
2500
2000
1500
1000
500
3.5
3
2.5
2
1.5
1
0.5
80
85
90
95
100
105
110
115
120
$\dot{Q}$ [Btu/s]
$\dot{m}_{steam}$ [lbm/s]
T_{steam} [F]


300
250
200
150
100
50
80
85
90
95
100
105
110
115
120
$\dot{m}_w$ [lbm/s]
T_{steam} [F]

11-61 Water is evaporated by hot exhaust gases in an evaporator. The rate of heat transfer, the exit temperature of the exhaust gases, and the rate of evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The heat of vaporization of water at 200°C is given to be h_{fg} = 1941 kJ/kg and specific heat of exhaust gases is given to be c_p = 1051 J/kg.°C.



Analysis The temperature differences between the water and the exhaust gases at the two ends of the evaporator are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 550°C - 200°C = 350°C$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = (T_{h,out} - 200)°C$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{350 - (T_{h,out} - 200)}{\ln\left[350/(T_{h,out} - 200)\right]}$$

Then the rate of heat transfer can be expressed as

$$\dot{Q} = UA_s\Delta T_{lm} = (1.780\ \text{kW/m}^2.°\text{C})(0.5\ \text{m}^2)\frac{350 - (T_{h,out} - 200)}{\ln\left[350/(T_{h,out} - 200)\right]} \qquad (1)$$

The rate of heat transfer can also be expressed as in the following forms

$$\dot{Q} = [\dot{m}c_p(T_{h,in} - T_{h,out})]_{\substack{\text{exhaust}\\ \text{gases}}} = (0.25\ \text{kg/s})(1.051\ \text{kJ/kg.°C})(550°\text{C} - T_{h,out}) \qquad (2)$$

$$\dot{Q} = (\dot{m}h_{fg})_{\text{water}} = \dot{m}_{\text{water}}(1941\ \text{kJ/kg}) \qquad (3)$$

We have three equations with three unknowns. Using an equation solver such as EES, the unknowns are determined to be

$$\dot{Q} = \mathbf{88.85\ kW}$$
$$T_{h,out} = \mathbf{211.8°C}$$
$$\dot{m}_{\text{water}} = \mathbf{0.0458\ kg/s}$$

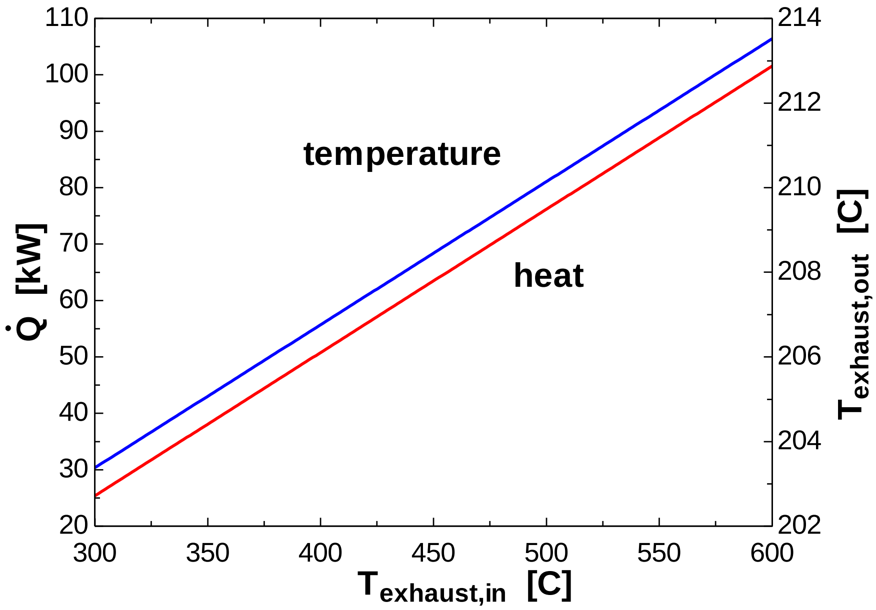
11-62 Prob. 11-61 is reconsidered. The effect of the exhaust gas inlet temperature on the rate of heat transfer, the exit temperature of exhaust gases, and the rate of evaporation of water is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

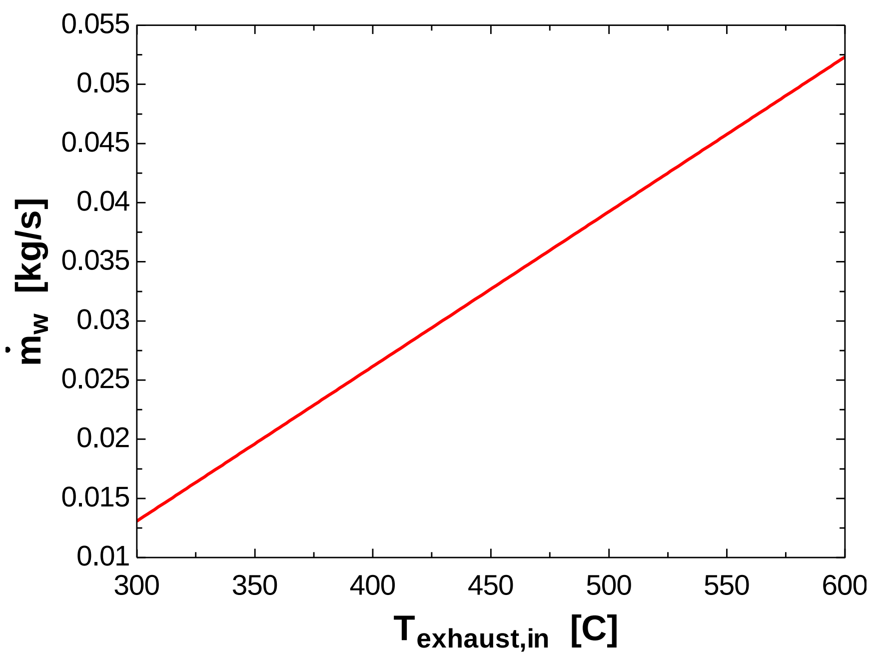
```
"GIVEN"
T_exhaust_in=550 [C]
c_p_exhaust=1.051 [kJ/kg-C]
m_dot_exhaust=0.25 [kg/s]
T_w=200 [C]
h_fg_w=1941 [kJ/kg]
A=0.5 [m^2]
U=1.780 [kW/m^2-C]

"ANALYSIS"
DELTAT_1=T_exhaust_in-T_w
DELTAT_2=T_exhaust_out-T_w
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_exhaust*c_p_exhaust*(T_exhaust_in-T_exhaust_out)
Q_dot=m_dot_w*h_fg_w
```

$T_{exhaust,in}$ [C]	$\dot{Q}$ [kW]	$T_{exhaust,out}$ [C]	$\dot{m}_w$ [kg/s]
300	25.39	203.4	0.01308
320	30.46	204.1	0.0157
340	35.54	204.7	0.01831
360	40.62	205.4	0.02093
380	45.7	206.1	0.02354
400	50.77	206.8	0.02616
420	55.85	207.4	0.02877
440	60.93	208.1	0.03139
460	66.01	208.8	0.03401
480	71.08	209.5	0.03662
500	76.16	210.1	0.03924
520	81.24	210.8	0.04185
540	86.32	211.5	0.04447
560	91.39	212.2	0.04709
580	96.47	212.8	0.0497
600	101.5	213.5	0.05232



110
100
90
80
70
60
50
40
30
20
214
212
210
208
206
204
202
300
350
400
450
500
550
600
temperature
heat
$\dot{Q}$ [kW]
$T_{exhaust,out}$ [C]
$T_{exhaust,in}$ [C]



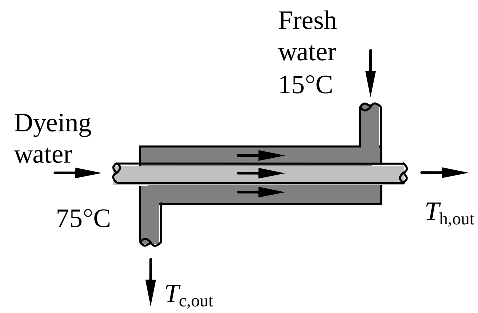
0.055
0.05
0.045
0.04
0.035
0.03
0.025
0.02
0.015
0.01
300
350
400
450
500
550
600
$\dot{m}_w$ [kg/s]
$T_{exhaust,in}$ [C]

11-63 The waste dyeing water is to be used to preheat fresh water. The outlet temperatures of each fluid and the mass flow rate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heats of waste dyeing water and the fresh water are given to be c_p = 4295 J/kg.°C and c_p = 4180 J/kg.°C, respectively.

Analysis The temperature differences between the dyeing water and the fresh water at the two ends of the heat exchanger are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 75 - T_{c,out}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = T_{h,out} - 15$$

and

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{(75 - T_{c,out}) - (T_{h,out} - 15)}{\ln\left[(75 - T_{c,out})/(T_{h,out} - 15)\right]}$$

Then the rate of heat transfer can be expressed as

$$\dot{Q} = UA_s \Delta T_{lm}$$
$$35\text{ kW} = (0.625\text{ kW/m}^2.°\text{C})(1.65\text{ m}^2)\frac{(75 - T_{c,out}) - (T_{h,out} - 15)}{\ln\left[(75 - T_{c,out})/(T_{h,out} - 15)\right]} \quad (1)$$

The rate of heat transfer can also be expressed as

$$\dot{Q} = [\dot{m}c_p(T_{h,in} - T_{h,out})]_{\substack{\text{dyeing}\\\text{water}}} \longrightarrow 35\text{ kW} = \dot{m}(4.295\text{ kJ/kg.°C})(75°\text{C} - T_{h,out}) \quad (2)$$

$$\dot{Q} = [\dot{m}c_p(T_{h,in} - T_{h,out})]_{\text{water}} \longrightarrow 35\text{ kW} = \dot{m}(4.18\text{ kJ/kg.°C})(T_{c,out} - 15°\text{C}) \quad (3)$$

We have three equations with three unknowns. Using an equation solver such as EES, the unknowns are determined to be

$$T_{c,out} = \mathbf{68.2°C}$$
$$T_{h,out} = \mathbf{23.3°C}$$
$$\dot{m} = \mathbf{0.158\ kg/s}$$

11-64 Counterflow double pipe heat exchanger with a surface area of 7.5 m^2 and U = 450 W/m^2·K is used to heat the engine oil using water at 100°C. It is to be determined if fouling has occurred in the heat exchanger over a period of time.

Assumptions **1** Steady state conditions exist. **2** Heat exchanger is well insulated. **3** Fluid properties remain constant.

Properties Heat capacity of engine oil is evaluated at an average temperature of (25 +55)°C/2 = 40°C from Table A-13: c_p = 1964 J/kg·K.

Analysis From the energy balance between hot water and engine oil we have,

Heat lost by water = Heat gained by engine oil

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = \dot{m}_h c_{ph}(T_{h,out} - T_{h,in})$$

$$\therefore \dot{Q} = (2.5\,\text{kg/s})(1964\,\text{J/kg}\cdot\text{K})(55 - 25)^\circ\text{C} = 147.3\ \text{kW}$$

Thus, the exit temperature of hot water is,

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{\dot{m}_h c_{ph}} = 100^o C - \frac{147.3\,\text{kW}}{(1.75\,\text{kg/s})(4206\,\text{J/kg}\cdot\text{K})} = 80\ ^\circ\text{C}$$

The heat transfer rate is calculated as

$$\dot{Q} = UA_s \Delta T_{lm}$$

Now the logarithmic temperature difference is calculated as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)},$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 100 - 55 = 45^o C$$

and

$$\Delta T_2 = T_{h,out} - T_{c,in} = 80 - 25 = 55^o C$$

$$\Delta T_{lm} = \frac{45 - 55}{\ln(45/55)} = 49.8\ ^\circ\text{C}$$

Therefore the actual overall heat transfer coefficient is

$$U = \frac{\dot{Q}}{A_s \Delta T_{lm}} = \frac{147.3 kW}{(7.5\,\text{m}^2)(49.8^\circ\text{C})} = 394.4\ \text{W/m}^2\cdot\text{K}$$

Since the actual overall heat transfer coefficient (394.4 W/m^2·K) is less than the designed value of the overall heat transfer coefficient (450 W/m^2·K), it can be concluded that fouling has occurred in the heat exchanger. The thermal resistance caused by fouling is then,

$$\frac{1}{U} = \frac{1}{U_{design}} + R_f \Rightarrow R_f = \frac{1}{U} - \frac{1}{U_{design}} = \frac{1}{394.4\,\text{W/m}^2\cdot\text{K}} - \frac{1}{450\,\text{W/m}^2\cdot\text{K}} = \mathbf{0.00031\ m^2\cdot K/W}$$

Discussion The fouling developed in the heat exchanger although cannot be prevented; it can be mitigated and controlled periodically using techniques such as chemical cleaning, reversal of flow direction and use of turbulence promoters.

11-65 Feed water at specified temperature and mass flow rate is to be heated by superheated steam in a counter flow heat exchanger. For the known values of convection heat transfer coefficient surface area of the heat exchanger in counter flow and parallel flow arrangement is to be determined.

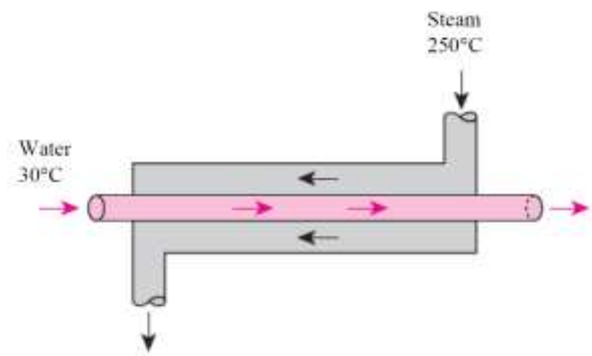
Assumptions **1** Steady state operating conditions exist. **2** Heat exchanger is well insulated. **3** Fouling on the steam side is assumed to be negligible. **4** Properties of steam and water stay constant. **5** Thermal resistance due to the pipe wall thickness is neglected.

Properties The specific heat of water is determined at an average temperature of (30 + 70)°C/2 = 50°C from Table A-9 to be c_p = 4181 J/kg·K.

Analysis For a 70°C drop in steam temperature, the steam exit temperature is $T_{h,out} = 250 - 70 = 180^oC$. The exit temperature of the water is to be maintained at a minimum temperature of 70°C.

(a) The rate of heat transfer in the heat exchanger is calculated as,

$$\dot{Q} = UA_s\Delta T_{lm} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$



where $$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 250 - 70 = 180^oC$$

and $$\Delta T_2 = T_{h,out} - T_{c,in} = 180 - 30 = 150^oC$$

$$\Delta T_{lm} = \frac{180 - 150}{\ln(180/150)} = 164.5^oC$$

Now the overall heat transfer coefficient (U) is determined as,

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + R_{f,i}$$

The fouling resistance on the feed water side is obtained from Table 11-2. Since the average temperature of the feed water as it flows through heat exchanger is 50°C, we take the average value of fouling factor above and below 50°C as 0.00015 m²·K/W.

$$\therefore \frac{1}{U} = \frac{1}{1250\,\text{W/m}^2\cdot\text{K}} + \frac{1}{850\text{W/m}^2\cdot\text{K}} + 0.00015\,\text{m}^2\cdot\text{K/W} \rightarrow U = 470.26 \;\; \text{W/m}^2\cdot\text{K}$$

The counter flow heat exchanger area to maintain the feed water exit temperature to a minimum of 70°C is calculated as,

$$A_s = \frac{\dot{m}_c C_{pc}(T_{c,out} - T_{c,in})}{U\Delta T_{lm}} = \frac{(3.47\,\text{kg/s})(4181\,\text{J/kg}\cdot\text{K})(70-30)^\circ\text{C}}{(470.26\,\text{W/m}^2\cdot\text{K})\,(164.5^\circ\text{C})} = 7.5 \;\; \text{m}^2$$

This is the heat exchanger area required to maintain the feed water exit temperature to a minimum of 70°C.

(b) In case of parallel flow arrangement, the log mean temperature difference is calculated as,

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}, \;\; \Delta T_1 = T_{h,in} - T_{c,in} = 250 - 30 = 220^oC$$

and

$$\Delta T_2 = T_{h,out} - T_{c,out} = 180 - 70 = 110^oC$$

$$\Delta T_{lm} = \frac{220 - 110}{\ln(220/110)} = 158.7^oC$$

Overall heat transfer coefficient remains the same for parallel flow arrangement. The parallel flow heat exchanger area to maintain the feed water exit temperature to a minimum of 70°C is calculated as,

$$A_s = \frac{\dot{m}_c c_{pc}(T_{c,out} - T_{c,in})}{U\Delta T_{lm}} = \frac{(3.47\,\text{kg/s})\,(4181\,\text{J/kg}\cdot\text{K})\,(70-30)^\circ\text{C}}{(470.26\,\text{W/m}^2\cdot\text{K})\,(158.7^\circ\text{C})} = 7.78\,\text{m}^2$$

Discussion The parallel flow arrangement requires about 3.6% higher heat transfer area. Fouling on the steam side is not considered in this problem. However, over a long run the superheated steam may cause fouling on the tubes outer surface leading to a decrease in the overall heat transfer coefficient and hence requiring a higher heat exchanger area.

11-66 During an experiment, the inlet and exit temperatures of water and oil and the mass flow rate of water are measured. The overall heat transfer coefficient based on the inner surface area is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

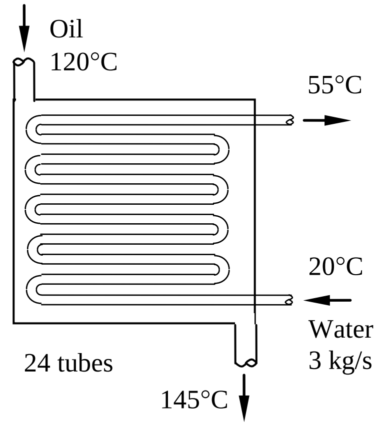
Properties The specific heats of water and oil are given to be 4180 and 2150 J/kg.°C, respectively.

Analysis The rate of heat transfer from the oil to the water is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (3\,\text{kg/s})(4.18\,\text{kJ/kg.°C})(55°\text{C} - 20°\text{C}) = 438.9\,\text{kW}$$

The heat transfer area on the tube side is

$$A_i = n\pi D_i L = 24\pi(0.012\,\text{m})(2\,\text{m}) = 1.8\,\text{m}^2$$



The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 120°\text{C} - 55°\text{C} = 65°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 45°\text{C} - 20°\text{C} = 25°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{65-25}{\ln(65/25)} = 41.9°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{55-20}{120-20} = 0.35 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{120-45}{55-20} = 2.14 \end{aligned}\right\} F = 0.70$$

Then the overall heat transfer coefficient becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{lm,CF}} = \frac{438.9\,\text{kW}}{(1.8\,\text{m}^2)(0.70)(41.9°\text{C})} = \mathbf{8.31\,kW/m^2.°C}$$

11-67 Oil is heated by water in a 1-shell pass and 6-tube passes heat exchanger. The rate of heat transfer and the heat transfer surface area are to be determined.

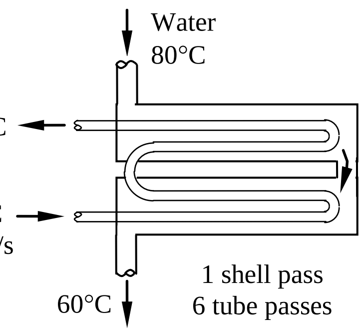
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heat of oil is given to be 2.0 kJ/kg.°C.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{oil}} = (10\text{ kg/s})(2.0\text{ kJ/kg.°C})(46°\text{C} - 25°\text{C}) = \mathbf{420\ kW}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 46°\text{C} = 34°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 25°\text{C} = 35°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{34 - 35}{\ln(34/35)} = 34.50°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{46 - 25}{80 - 25} = 0.38 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80 - 60}{46 - 25} = 0.95 \end{aligned}\right\} F = 0.93$$

Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm,CF}} = \frac{420\text{ kW}}{(1.0\text{ kW/m}^2.°\text{C})(0.93)(34.50°\text{C})} = \mathbf{13.1\ m^2}$$

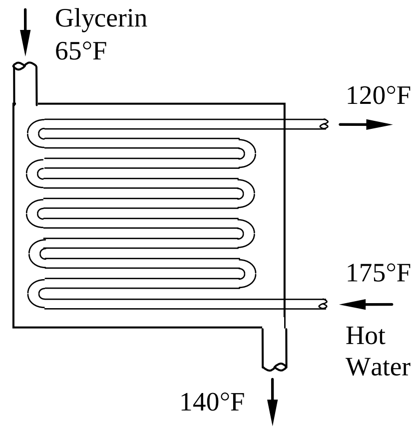
11-68E Glycerin is heated by hot water in a 1-shell pass and 8-tube passes heat exchanger. The rate of heat transfer for the cases of fouling and no fouling are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Heat transfer coefficients and fouling factors are constant and uniform. **5** The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

Properties The specific heats of glycerin and water are given to be 0.60 and 1.0 Btu/lbm.°F, respectively.

Analysis (*a*) The tubes are thin walled and thus we assume the inner surface area of the tube to be equal to the outer surface area. Then the heat transfer surface area of this heat exchanger becomes

$$A_s = n\pi DL = 8\pi(0.5/12\text{ ft})(500\text{ ft}) = 523.6\text{ ft}^2$$



The temperature differences at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 175°\text{F} - 140°\text{F} = 35°\text{F}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 120°\text{F} - 65°\text{F} = 55°\text{F}$$

and $$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{35-55}{\ln(35/55)} = 44.25°\text{F}$$

The correction factor is

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{120-175}{65-175} = 0.50 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{65-140}{120-175} = 1.36 \end{aligned}\right\} F = 0.50$$

In case of no fouling, the overall heat transfer coefficient is determined from

$$U = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{50\text{ Btu/h.ft}^2.°\text{F}} + \dfrac{1}{4\text{ Btu/h.ft}^2.°\text{F}}} = 3.704\text{ Btu/h.ft}^2.°\text{F}$$

Then the rate of heat transfer becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} = (3.704\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)(0.50)(44.25°\text{F}) = \mathbf{42{,}910\text{ Btu/h}}$$

(*b*) The thermal resistance of the heat exchanger with a fouling factor is

$$\begin{aligned} R &= \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + \frac{1}{h_o A_o} \\ &= \frac{1}{(50\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)} + \frac{0.002\text{ h.ft}^2.°\text{F/Btu}}{523.6\text{ ft}^2} + \frac{1}{(4\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)} \\ &= 0.0005195\text{ h.°F/Btu} \end{aligned}$$

The overall heat transfer coefficient in this case is

$$R = \frac{1}{UA_s} \longrightarrow U = \frac{1}{RA_s} = \frac{1}{(0.0005195\text{ h.°F/Btu})(523.6\text{ ft}^2)} = 3.676\text{ Btu/h.ft}^2.°\text{F}$$

Then rate of heat transfer becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} = (3.676\text{ Btu/h.ft}^2.°\text{F})(523.6\text{ ft}^2)(0.50)(44.25°\text{F}) = \mathbf{42{,}585\text{ Btu/h}}$$

11-69 Water is heated by ethylene glycol in a 2-shell passes and 12-tube passes heat exchanger. The rate of heat transfer and the heat transfer surface area on the tube side are to be determined.

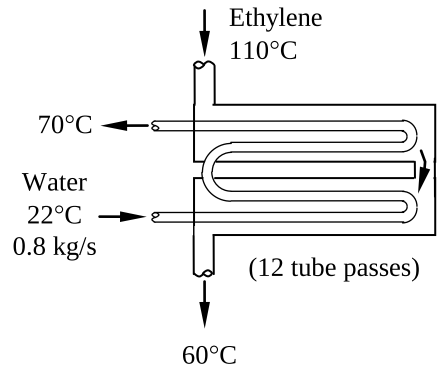
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heats of water and ethylene glycol are given to be 4.18 and 2.68 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is :

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (0.8\,\text{kg/s})(4.18\,\text{kJ/kg.°C})(70°\text{C} - 22°\text{C}) = \mathbf{160.5\,kW}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 110°\text{C} - 70°\text{C} = 40°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 22°\text{C} = 38°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{40 - 38}{\ln(40/38)} = 39°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 22}{110 - 22} = 0.55 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{110 - 60}{70 - 22} = 1.04 \end{aligned}\right\} F = 0.92$$

Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{160.5\,\text{kW}}{(0.28\,\text{kW/m}^2.°\text{C})(0.92)(39°\text{C})} = \mathbf{16.0\,m^2}$$

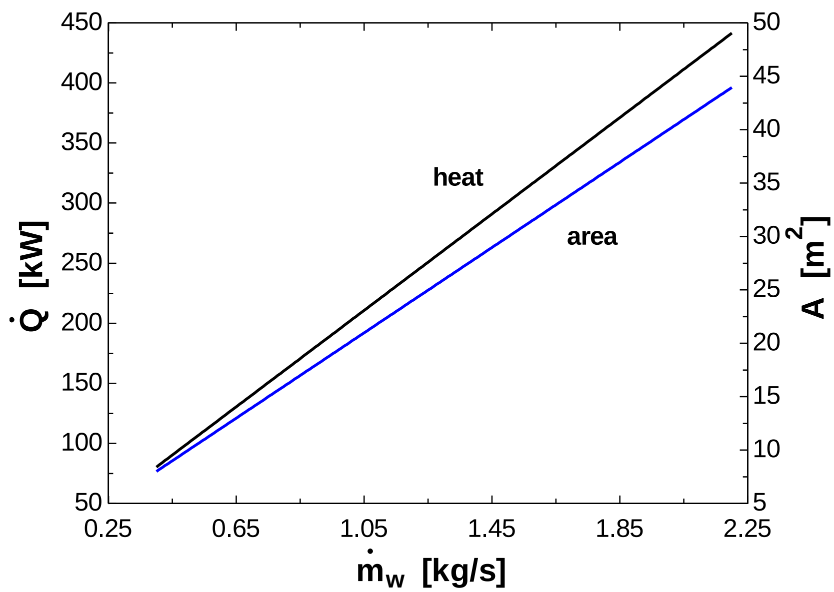
11-70 Prob. 11-69 is reconsidered. The effect of the mass flow rate of water on the rate of heat transfer and the tube-side surface area is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

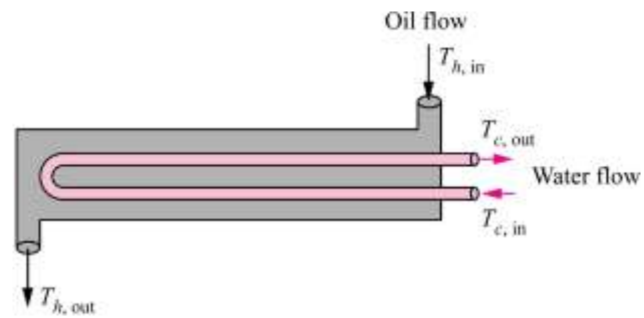
```
"GIVEN"
T_w_in=22 [C]
T_w_out=70 [C]
m_dot_w=0.8 [kg/s]
c_p_w=4.18 [kJ/kg-C]
T_glycol_in=110 [C]
T_glycol_out=60 [C]
c_p_glycol=2.68 [kJ/kg-C]
U=0.28 [kW/m^2-C]

"ANALYSIS"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
Q_dot=m_dot_glycol*c_p_glycol*(T_glycol_in-T_glycol_out)
DELTAT_1=T_glycol_in-T_w_out
DELTAT_2=T_glycol_out-T_w_in
DELTAT_lm_CF=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
P=(T_w_out-T_w_in)/(T_glycol_in-T_w_in)
R=(T_glycol_in-T_glycol_out)/(T_w_out-T_w_in)
F=0.92 "from Fig. 11-19b of the text at the calculated P and R"
Q_dot=U*A*F*DELTAT_lm_CF
```

$\dot{m}_w$ [kg/s]	$\dot{Q}$ [kW]	A [m^2]
0.4	80.26	7.99
0.5	100.3	9.988
0.6	120.4	11.99
0.7	140.4	13.98
0.8	160.5	15.98
0.9	180.6	17.98
1	200.6	19.98
1.1	220.7	21.97
1.2	240.8	23.97
1.3	260.8	25.97
1.4	280.9	27.97
1.5	301	29.96
1.6	321	31.96
1.7	341.1	33.96
1.8	361.2	35.96
1.9	381.2	37.95
2	401.3	39.95
2.1	421.3	41.95
2.2	441.4	43.95



11-71E A 1-shell and 2-tube heat exchanger has specified overall heat transfer coefficient, inlet and outlet temperatures, and mass flow rates, (*a*) the log mean temperature difference and (*b*) the surface area of the heat exchanger are to be determined.



Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of water is given to be c_{pc} = 1.0 Btu/lbm·°F.

Analysis (*a*) Using Fig. 11-19*a*, the correction factor can be determined to be

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{100-80}{180-80} = 0.2 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{180-120}{100-80} = 3.0 \end{aligned}\right\} F \approx 0.94 \quad \text{(Fig. 11-19}a\text{)}$$

The log mean temperature difference for the counter-flow arrangement is

$$\Delta T_{\text{lm, CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(180-100)-(120-80)}{\ln[(180-100)/(120-80)]}\ \text{°C} = 57.7\text{°F}$$

Hence, the log mean temperature difference is

$$\Delta T_{\text{lm,}} = F\Delta T_{\text{lm, CF}} = 0.94(57.7\text{°F}) = \mathbf{54.2°F}$$

(*b*) The surface area of the heat exchanger can be determined using

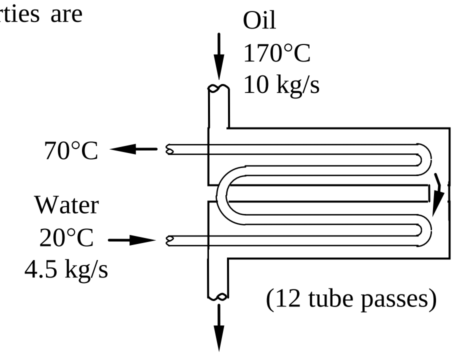
$$\dot{Q} = UA_s F\Delta T_{\text{lm, CF}} \quad \longrightarrow \quad A_s = \frac{\dot{Q}}{UF\Delta T_{\text{lm, CF}}} = \frac{\dot{m}_c c_{pc}(T_{c,\text{out}} - T_{c,\text{in}})}{UF\Delta T_{\text{lm, CF}}}$$

$$A_s = \frac{(20{,}000\ \text{lbm/hr})(1.0\ \text{Btu/lbm}\cdot\text{°F})(100-80)\text{°F}}{(40\ \text{Btu/hr}\cdot\text{ft}^2\cdot\text{°F})(0.94)(57.7\text{°F})} = \mathbf{184\ ft^2}$$

Discussion The surface area of the heat exchanger can also be determined using the effectiveness-NTU method.

11-72 Water is heated by hot oil in a 2-shell passes and 12-tube passes heat exchanger. The heat transfer surface area on the tube side is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.



Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} \\ &= (4.5\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) \\ &= 940.5\text{ kW}\end{aligned}$$

The outlet temperature of the oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = 129°\text{C}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\begin{aligned}\Delta T_1 &= T_{h,in} - T_{c,out} = 170°\text{C} - 70°\text{C} = 100°\text{C} \\ \Delta T_2 &= T_{h,out} - T_{c,in} = 129°\text{C} - 20°\text{C} = 109°\text{C}\end{aligned}$$

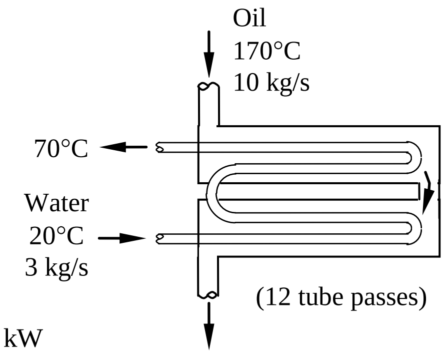
$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{100 - 109}{\ln(100/109)} = 104.4°\text{C}$$

$$\left.\begin{aligned}P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 20}{170 - 20} = 0.33 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{170 - 129}{70 - 20} = 0.82\end{aligned}\right\} F = 1.0$$

Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = UA_s F\Delta T_{lm,CF} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm,CF}} = \frac{940.5\text{ kW}}{(0.350\text{ kW/m}^2.°\text{C})(1.0)(104.4°\text{C})} = \mathbf{25.7\ m^2}$$

11-73 Water is heated by hot oil in a 2-shell passes and 12-tube passes heat exchanger. The heat transfer surface area on the tube side is to be determined.



Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (3\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = 627\text{ kW}$$

The outlet temperature of the oil is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{oil}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{627\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = 142.7°\text{C}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 170°\text{C} - 70°\text{C} = 100°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 142.7°\text{C} - 20°\text{C} = 122.7°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{100 - 122.7}{\ln(100/122.7)} = 111.0°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70-20}{170-20} = 0.33 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{170-142.7}{70-20} = 0.55 \end{aligned}\right\} F = 1.0$$

Then the heat transfer surface area on the tube side becomes

$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{627\text{ kW}}{(0.350\text{ kW/m}^2.°\text{C})(1.0)(111.0°\text{C})} = \mathbf{16.1\,m^2}$$

11-74 Ethyl alcohol is heated by water in a 2-shell passes and 8-tube passes heat exchanger. The heat transfer surface area of the heat exchanger is to be determined.

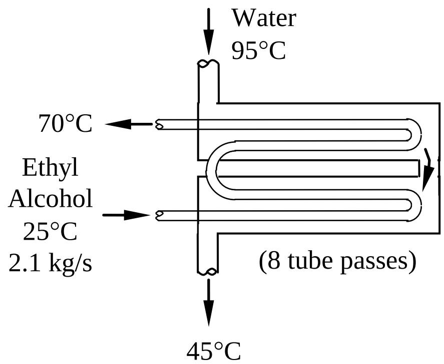
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** There is no fouling. **5** Fluid properties are constant.

Properties The specific heats of water and ethyl alcohol are given to be 4.19 and 2.67 kJ/kg.°C, respectively.

Analysis The rate of heat transfer in this heat exchanger is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{ethylalcohol}} = (2.1\,\text{kg/s})(2.67\,\text{kJ/kg.°C})(70°\text{C} - 25°\text{C}) = 252.3\,\text{kW}$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 95°\text{C} - 70°\text{C} = 25°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 45°\text{C} - 25°\text{C} = 20°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25 - 20}{\ln(25/20)} = 22.4°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 25}{95 - 25} = 0.64 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{95 - 45}{70 - 25} = 1.1 \end{aligned}\right\} F = 0.82$$

Then the heat transfer surface area on the tube side becomes

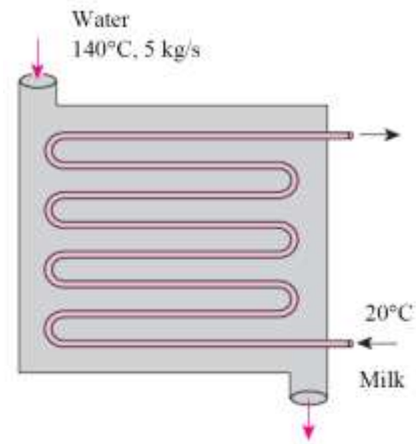
$$\dot{Q} = U_i A_i F \Delta T_{lm,CF} \longrightarrow A_i = \frac{\dot{Q}}{U_i F \Delta T_{lm,CF}} = \frac{252.3\,\text{kW}}{(0.950\,\text{kW/m}^2.°\text{C})(0.82)(22.4°\text{C})} = \mathbf{14.5\,m^2}$$

11-75 One shell and 10 tube pass heat exchanger is used in a milk pasteurizing process. The width of the shell (tube length in each pass) is to be determined in order to ensure complete pasteurization of milk.

Assumptions **1** Steady state conditions exist. **2** Thermal properties of milk and hot water remain constant. **3** heat exchanger is well insulated. **4** No fouling inside the heat exchanger.

Properties The specific heat of water is evaluated from Table A-9 at an average temperature of (140 + 80)°C/2 = 110 °C. $c_{ph} = 4229\ \text{J/kg}\cdot\text{K}$

Analysis From energy balance between the cold milk and hot water we have



Heat lost by water = Heat gained by milk.

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\dot{Q} = (5\ \text{kg/s})(4229\ \text{J/kg}\cdot\text{K})(140 - 80)^{\circ}\text{C} = 1268.7\ \text{kW}$$

The actual rate of heat transfer can also be expressed as

$$\dot{Q} = UA_s F\Delta T_{lm,CF}$$

The overall heat transfer coefficient is determined from the convective heat transfer coefficient values on milk and water side.

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} = \frac{1}{450\ \text{W/m}^2\cdot\text{K}} + \frac{1}{1100\ \text{W/m}^2\cdot\text{K}} \rightarrow U = 319.35\ \text{W/m}^2\cdot\text{K}$$

The log mean temperature difference is determined as

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 140 - 70 = 70^{o}C$$

and $$\Delta T_2 = T_{h,out} - T_{c,in} = 80 - 20 = 60^{o}C$$

$$\Delta T_{lm,CF} = \frac{70 - 60}{\ln(70/60)} = 64.87^{o}C$$

Since this is a multiple pass heat exchanger we need to determine the correction factor F from the temperature ratios P and R.

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{70 - 20}{140 - 20} = 0.417 \qquad \text{and} \qquad R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{140 - 80}{70 - 20} = 1.2$$

From Figure 11-19(a), for the corresponding values of P and R we get,

$F = 0.865$

Thus the actual rate of heat transfer is,

$$\dot{Q} = (319.35\ \text{W/m}^2\cdot\text{K})(A_s m^2)(0.865)(64.87^{o}\text{C})$$

$$\therefore A_s = \frac{1268.7\times 10^3\ \text{W}}{(319.35\text{W/m}^2\cdot\text{K})(0.865)(64.87^{o}C)} = 70.79\ \text{m}^2$$

The surface area of the heat exchanger tubes is,

$$A_s = \pi DL \times n \times p$$

where ‘n’ and ‘p’ are the number of tubes and tube passes, respectively. Therefore the length of each tube is,

$$L = \frac{70.79\text{m}^2}{\pi(0.02\ \text{m})(30)(10)} = \mathbf{3.75\ m}$$

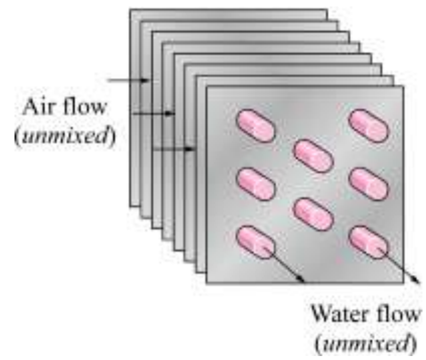
Discussion In this problem it is assumed that the effect of fouling on the overall heat transfer coefficient is negligible. However, considering that the fouling will deteriorate the performance of the heat exchanger over a period of time, a suitable factor of safety must be considered in deciding upon the length of heat exchanger.

11-76E A single-pass cross-flow heat exchanger is used to cool jacket water using air. The log mean temperature difference for the heat exchanger is to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heats of both water and air are given to be c_{ph} = 1.0 Btu/lbm·°F and c_{pc} = 0.245 Btu/lbm·°F, respectively.

Analysis The rate of heat transfer in the heat exchanger is



$$\begin{aligned}\dot{Q} &= \dot{m}_h c_{ph}(T_{h,\mathrm{in}} - T_{h,\mathrm{out}}) \\ &= (92{,}000\ \mathrm{lbm/hr})(1.0\ \mathrm{Btu/lbm\cdot°F})(190-140)\mathrm{°F} \\ &= 4.6\times10^6\ \mathrm{Btu/hr}\end{aligned}$$

Since heat transfer from the hot fluid is equal to the heat transfer to the cold fluid, we have

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,\mathrm{out}} - T_{c,\mathrm{in}}) \quad\rightarrow\quad T_{c,\mathrm{out}} = \frac{\dot{Q}}{\dot{m}_c c_{pc}} + T_{c,\mathrm{in}}$$

$$T_{c,\mathrm{out}} = \frac{4.6\times10^6\ \mathrm{Btu/hr}}{(400{,}000\ \mathrm{lbm/hr})(0.245\ \mathrm{Btu/lbm\cdot°F})} + 90\mathrm{°F} = 136.9\mathrm{°F}$$

Thus, the log mean temperature difference for the counter-flow arrangement is

$$\Delta T_{\mathrm{lm,\,CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(190-136.9)-(140-90)}{\ln[(190-136.9)/(140-90)]}\ \mathrm{°F} = 51.5\mathrm{°F}$$

Using Fig. 11-19*c*, the correction factor can be determined to be

$$\left.\begin{aligned}P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{140-190}{90-190} = 0.50 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{90-136.9}{140-190} = 0.94\end{aligned}\right\}\ F \approx 0.92 \qquad \text{(Fig. 11-19}c\text{)}$$

The log mean temperature difference is

$$\Delta T_{\mathrm{lm,}} = F\Delta T_{\mathrm{lm,\,CF}} = 0.92(51.5\mathrm{°F}) = \mathbf{47.4°F}$$

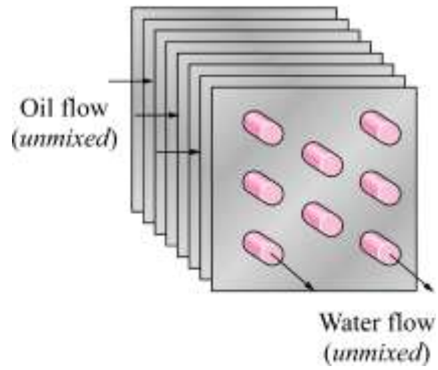
Discussion The correction factor (F) represents how closely the cross-flow heat exchanger approximates a counter-flow heat exchanger in terms of its logarithmic mean temperature difference.

11-77 A single-pass cross-flow heat exchanger with both fluids unmixed, the value of the overall heat transfer coefficient is to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The properties of oil are given to be c_{ph} = 1.93 kJ/kg·K and ρ = 870 kg/m^3.

Analysis The mass flow rate of oil (hot fluid) is



$$\dot{m}_h = \rho\dot{\boldsymbol{V}} = (870\,\text{kg/m}^3)(0.19\,\text{m}^3/\text{min})(1/60\,\text{min/s}) = 2.755\,\text{kg/s}$$

Using energy balance on the hot fluid, we have

$$\begin{aligned}\dot{Q} &= \dot{m}_h c_{ph}(T_{h,\text{in}} - T_{h,\text{out}})\\ &= (2.755\,\text{kg/s})(1930\,\text{J/kg}\cdot\text{K})(38-29)\,\text{K}\\ &= 4.785\times10^4\ \text{W}\end{aligned}$$

Using Fig. 11-19*c*, the correction factor can be determined to be

$$\left.\begin{aligned}P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{33-16}{38-16} = 0.77\\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{38-29}{33-16} = 0.53\end{aligned}\right\}\ F \approx 0.85 \qquad \text{(Fig. 11-19}c\text{)}$$

The log mean temperature difference for the counter-flow arrangement is

$$\Delta T_{\text{lm, CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(38-33)-(29-16)}{\ln[(38-33)/(29-16)]}\,°\text{C} = 8.372°\text{C}$$

Thus, the overall heat transfer coefficient can be determined using

$$\dot{Q} = UA_s F\Delta T_{\text{lm, CF}} \qquad \rightarrow \qquad U = \frac{\dot{Q}}{A_s F\Delta T_{\text{lm, CF}}}$$

$$U = \frac{4.785\times10^4\ \text{W}}{(20\,\text{m}^2)(0.85)(8.372\,\text{K})} = \mathbf{336\ W/m^2\cdot K}$$

Discussion Cross-flow heat exchangers are commonly found in automobile radiators.

The Effectiveness-NTU Method

11-78C The effectiveness of a heat exchanger is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate and represents how closely the heat transfer in the heat exchanger approaches to maximum possible heat transfer. Since the actual heat transfer rate can not be greater than maximum possible heat transfer rate, the effectiveness can not be greater than one. The effectiveness of a heat exchanger depends on the geometry of the heat exchanger as well as the flow arrangement.

11-79C For a specified fluid pair, inlet temperatures and mass flow rates, the counter-flow heat exchanger will have the highest effectiveness.

11-80C Once the effectiveness ε is known, the rate of heat transfer and the outlet temperatures of cold and hot fluids in a heat exchanger are determined from

$$\dot{Q} = \varepsilon\dot{Q}_{\max} = \varepsilon C_{\min}(T_{h,in} - T_{c,in})$$
$$\dot{Q} = \dot{m}_c c_{p,c}(T_{c,out} - T_{c,in})$$
$$\dot{Q} = \dot{m}_h c_{p,h}(T_{h,in} - T_{h,out})$$

11-81C The heat transfer in a heat exchanger will reach its maximum value when the hot fluid is cooled to the inlet temperature of the cold fluid. Therefore, the temperature of the hot fluid cannot drop below the inlet temperature of the cold fluid at any location in a heat exchanger.

11-82C The heat transfer in a heat exchanger will reach its maximum value when the cold fluid is heated to the inlet temperature of the hot fluid. Therefore, the temperature of the cold fluid cannot rise above the inlet temperature of the hot fluid at any location in a heat exchanger.

11-83C The fluid with the lower mass flow rate will experience a larger temperature change. This is clear from the relation

$$\dot{Q} = \dot{m}_c c_p \Delta T_{cold} = \dot{m}_h c_p \Delta T_{hot}$$

11-84C The maximum possible heat transfer rate is in a heat exchanger is determined from

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in})$$

where $C_{\min}$ is the smaller heat capacity rate. The value of $\dot{Q}_{\max}$ does not depend on the type of heat exchanger.

11-85C When the capacity ratio is equal to zero and the number of transfer units value is greater than 5, a counter-flow heat exchanger has an effectiveness of one. In this case the exit temperature of the fluid with smaller capacity rate will equal to inlet temperature of the other fluid. For a parallel-flow heat exchanger the answer would be the same.

11-86C The increase of effectiveness with NTU is not linear. The effectiveness increases rapidly with NTU for small values (up to abo ut NTU = 1.5), but rather slowly for larger values. Therefore, the effectiveness will not double when the length of heat exchanger is doubled.

11-87C A heat exchanger has the smallest effectiveness value when the heat capacity rates of two fluids are identical. Therefore, reducing the mass flow rate of cold fluid by half will increase its effectiveness.

11-88C The longer heat exchanger is more likely to have a higher effectiveness.

11-89C The NTU of a heat exchanger is defined as $NTU = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}c_p)_{\min}}$ where U is the overall heat transfer coefficient and A_s is the heat transfer surface area of the heat exchanger. For specified values of U and $C_{\min}$, the value of NTU is a measure of the heat exchanger surface area A_s. Because the effectiveness increases slowly for larger values of NTU, a large heat exchanger cannot be justified economically. Therefore, a heat exchanger with a very large NTU is not necessarily a good one to buy.

11-90C The value of effectiveness increases slowly with a large values of NTU (usually larger than 3). Therefore, doubling the size of the heat exchanger will not save much energy in this case since the increase in the effectiveness will be very small.

11-91C The value of effectiveness increases rapidly with small values of NTU (up to about 1.5). Therefore, tripling the NTU will cause a rapid increase in the effectiveness of the heat exchanger, and thus saves energy. I would support this proposal.

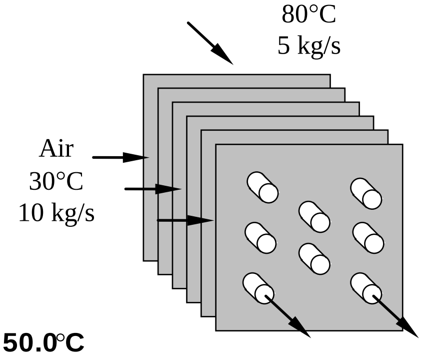
11-92 Hot water coming from the engine of an automobile is cooled by air in the radiator. The outlet temperature of the air and the rate of heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and air are given to be 4.00 and 1.00 kJ/kg.°C, respectively.

Analysis (*a*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (5\text{ kg/s})(4.00\text{ kJ/kg.°C}) = 20\text{ kW/°C}$$
$$C_c = \dot{m}_c c_{pc} = (10\text{ kg/s})(1.00\text{ kJ/kg.°C}) = 10\text{ kW/°C}$$



Therefore

$$C_{\min} = C_c = 10\text{ kW/°C}$$

which is the smaller of the two heat capacity rates. Noting that the heat capacity rate of the air is the smaller one, the outlet temperature of the air is determined from the effectiveness relation to be

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_{\min}(T_{a,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} \longrightarrow 0.4 = \frac{(T_{a,out} - 30)\text{°C}}{(80-30)\text{°C}} \longrightarrow T_{a,out} = \mathbf{50.0°C}$$

(*b*) The rate of heat transfer is determined from

$$\dot{Q} = C_{air}(T_{a,out} - T_{a,in}) = (10\text{ kW/°C})(50\text{°C} - 30\text{°C}) = \mathbf{200\ kW}$$

11-93 Inlet and outlet temperatures of the hot and cold fluids in a double-pipe heat exchanger are given. It is to be determined whether this is a parallel-flow or counter-flow heat exchanger and the effectiveness of it.

Analysis This is a counter-flow heat exchanger because in the parallel-flow heat exchangers the outlet temperature of the cold fluid (55°C in this case) cannot exceed the outlet temperature of the hot fluid, which is (45°C in this case). Noting that the mass flow rates of both hot and cold oil streams are the same, we have $C_{\min} = C_{\max}$. Then the effectiveness of this heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_h(T_{h,in} - T_{c,in})} = \frac{80\text{°C} - 45\text{°C}}{80\text{°C} - 20\text{°C}} = \mathbf{0.583}$$

11-94E Inlet and outlet temperatures of the hot and cold fluids in a double-pipe heat exchanger are given. It is to be determined the fluid, which has the smaller heat capacity rate and the effectiveness of the heat exchanger.

Analysis Hot water has the smaller heat capacity rate since it experiences a greater temperature change. The effectiveness of this heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{C_h(T_{h,in} - T_{h,out})}{C_h(T_{h,in} - T_{c,in})} = \frac{190\text{°F} - 100\text{°F}}{190\text{°F} - 70\text{°F}} = \mathbf{0.75}$$

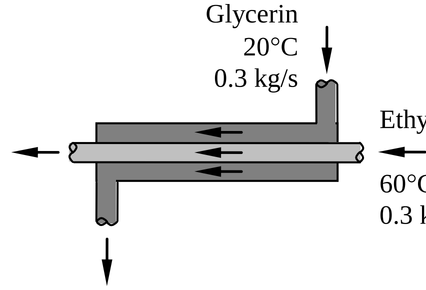
11-95 Glycerin is heated by ethylene glycol in a heat exchanger. Mass flow rates and inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

Properties The specific heats of the glycerin and ethylene glycol are given to be 2.4 and 2.5 kJ/kg.°C, respectively.

Analysis (*a*) The heat capacity rates of the hot and cold fluids are

$$C_c = \dot{m}_c c_{pc} = (0.3\,\text{kg/s})(2400\,\text{J/kg.°C}) = 720\,\text{W/°C}$$
$$C_h = \dot{m}_h c_{ph} = (0.3\,\text{kg/s})(2500\,\text{J/kg.°C}) = 750\,\text{W/°C}$$



Therefore,

$$C_{\min} = C_c = 720\,\text{W/°C}$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{720}{750} = 0.96$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (720\text{ W/°C})(60°\text{C} - 20°\text{C}) = 28{,}800\text{ W} = 28.8\,\text{kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(380\text{ W/m}^2.°\text{C})(5.3\,\text{m}^2)}{720\text{ W/°C}} = 2.797$$

Effectiveness of this heat exchanger corresponding to c = 0.96 and NTU = 2.058 is determined using the proper relation in Table 11-4

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-2.797(1+0.96)]}{1+0.96} = 0.5081$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.5081)(28.8\,\text{kW}) = \mathbf{14.63kW}$$

(*b*) Finally, the outlet temperatures of the cold and the hot fluid streams are determined from

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20°\text{C} + \frac{14.63\,\text{kW}}{0.720\,\text{kW/°C}} = \mathbf{40.3°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 60°\text{C} - \frac{14.63\,\text{kW}}{0.750\,\text{kW/°C}} = \mathbf{40.5°C}$$

11-96 A thin-walled concentric tube counter-flow heat exchanger has specified mass flow rates and inlet temperatures, (*a*) the heat transfer rate for the heat exchanger, (*b*) the outlet temperatures of the cold and hot fluids, and (*c*) the fouling factor after a period of operation are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heats of the hot and cold fluids are given to be c_{ph} = 4188 J/kg·K and c_{pc} = 4178 J/kg·K, respectively.

Analysis (*a*) The heat capacity rates are

$$C_c = \dot{m}_c c_{pc} = (5\text{ kg/s})(4178\text{ J/kg}\cdot\text{K}) = 20890\text{ W/K}$$

$$C_h = \dot{m}_h c_{ph} = (2.5\text{ kg/s})(4188\text{ J/kg}\cdot\text{K}) = 10470\text{ W/K}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{C_h}{C_c} = \frac{10470\text{ W/K}}{20890\text{ W/K}} = 0.5012$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(1000\text{ W/m}^2\cdot\text{K})(23\text{ m}^2)}{10470\text{ W/K}} = 2.197$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\varepsilon = \frac{1-\exp[-\text{NTU}(1-c)]}{1-c\exp[-\text{NTU}(1-c)]} = \frac{1-\exp[-2.197(1-0.5012)]}{1-(0.5012)\exp[-2.197(1-0.5012)]} = 0.7997$$

The heat transfer rate for the heat exchanger is

$$\dot{Q} = C_{\min}\varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) = (10470\text{ W/K})(0.7997)(100-20)\text{ K} = \mathbf{6.70\times10^5\ W}$$

(*b*) The outlet temperatures of the cold and hot fluids are

$$\dot{Q} = C_c(T_{c,\text{out}} - T_{c,\text{in}}) \quad\rightarrow\quad T_{c,\text{out}} = \frac{\dot{Q}}{C_c} + T_{c,\text{in}} = \frac{6.70\times10^5\text{ W}}{20890\text{ W/K}} + 20°\text{C} = \mathbf{52.1°C}$$

and

$$\dot{Q} = C_h(T_{h,\text{in}} - T_{h,\text{out}}) \quad\rightarrow\quad T_{h,\text{out}} = T_{h,\text{in}} - \frac{\dot{Q}}{C_h} = 100°\text{C} - \frac{6.70\times10^5\text{ W}}{10470\text{ W/K}} = \mathbf{36.0°C}$$

(*c*) The overall heat transfer coefficient at clean conditions is U_{clean} = 1000 W/m^2·K. After a period of operation, the overall heat transfer coefficient is reduced to U_{dirty} = 500 W/m^2·K. Hence, the fouling factor can be determined to be

$$\frac{1}{U_{\text{dirty}}} = \frac{1}{U_{\text{clean}}} + R_f \quad\rightarrow\quad R_f = \frac{1}{U_{\text{dirty}}} - \frac{1}{U_{\text{clean}}}$$

$$R_f = \left(\frac{1}{500} - \frac{1}{1000}\right)\text{m}^2\cdot\text{K/W} = \mathbf{0.001\,m^2\cdot K/W}$$

Discussion Using Figure 11-27*b*, the heat transfer effectiveness is approximately $\varepsilon \approx 78\%$.

11-97 Water is heated by solar-heated hot air in a heat exchanger. The mass flow rates and the inlet temperatures are given. The outlet temperatures of the water and the air are to be determined.

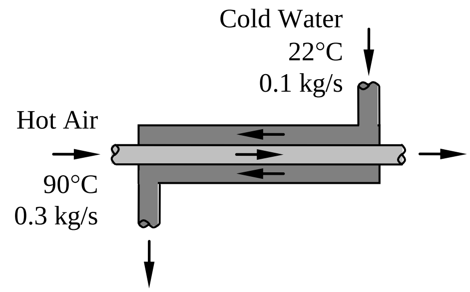
Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg.°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.3\text{kg/s})(1010\text{ J/kg.°C}) = 303\text{ W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.1\text{kg/s})(4180\text{ J/kg.°C}) = 418\text{ W/°C}$$



Therefore,

$$C_{\min} = C_c = 303\text{ W/°C}$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{303}{418} = 0.725$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (303\text{ W/°C})(90°\text{C} - 22°\text{C}) = 20{,}604\text{ kW}$$

The heat transfer surface area is

$$A_s = \pi DL = (\pi)(0.012\text{ m})(12\text{ m}) = 0.45\text{ m}^2$$

Then the NTU of this heat exchanger becomes

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(80\text{ W/m}^2.°\text{C})(0.45\text{ m}^2)}{303\text{ W/°C}} = 0.119$$

The effectiveness of this counter-flow heat exchanger corresponding to c = 0.725 and NTU = 0.119 is determined using the relation in Table 11-4 to be

$$\varepsilon = \frac{1-\exp[-NTU(1-c)]}{1-c\exp[-NTU(1-c)]} = \frac{1-\exp[-0.119(1-0.725)]}{1-0.725\exp[-0.119(1-0.725)]} = 0.108$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon\dot{Q}_{\max} = (0.108)(20{,}604\text{ W}) = 2225.2\text{ W}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 22°\text{C} + \frac{2225.2\text{ W}}{418\text{ W / °C}} = \mathbf{27.3°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 90°\text{C} - \frac{2225.2\text{ W}}{303\text{ W/°C}} = \mathbf{82.7°C}$$

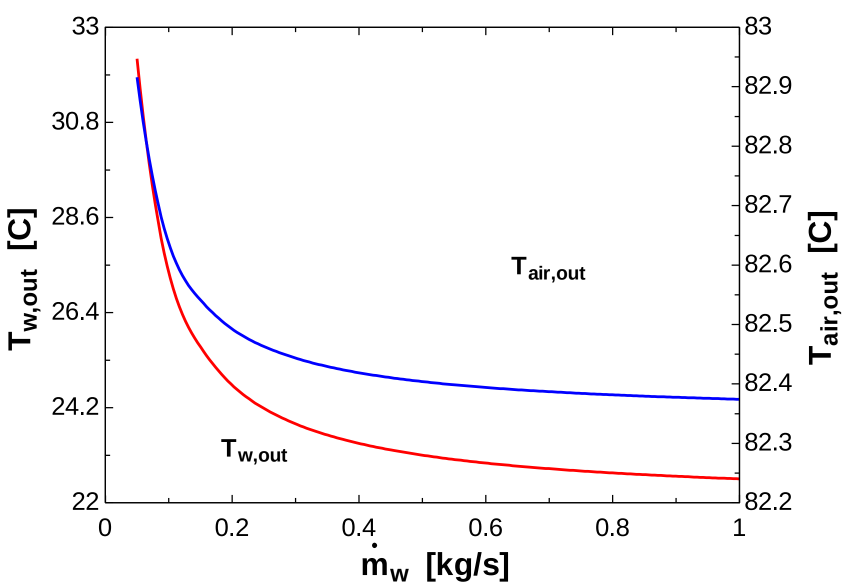
11-98 Prob. 11-97 is reconsidered. The effects of the mass flow rate of water and the tube length on the outlet temperatures of water and air are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

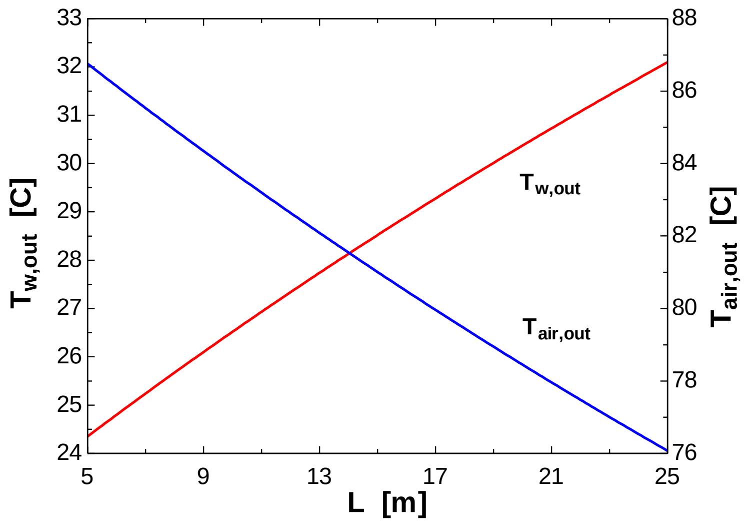
```
"GIVEN"
T_air_in=90 [C]
m_dot_air=0.3 [kg/s]
c_p_air=1.01 [kJ/kg-C]
T_w_in=22 [C]
m_dot_w=0.1 [kg/s]
c_p_w=4.18 [kJ/kg-C]
U=0.080 [kW/m^2-C]
L=12 [m]
D=0.012 [m]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use LMTD method. Both methods give the same results."
DELTAT_1=T_air_in-T_w_out
DELTAT_2=T_air_out-T_w_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
A=pi*D*L
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_air*c_p_air*(T_air_in-T_air_out)
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
```

$\dot{m}_w$ [kg/s]	$T_{w,out}$ [C]	$T_{air,out}$ [C]
0.05	32.27	82.92
0.1	27.34	82.64
0.15	25.6	82.54
0.2	24.72	82.49
0.25	24.19	82.46
0.3	23.83	82.44
0.35	23.57	82.43
0.4	23.37	82.42
0.45	23.22	82.41
0.5	23.1	82.4
0.55	23	82.4
0.6	22.92	82.39
0.65	22.85	82.39
0.7	22.79	82.39
0.75	22.74	82.38
0.8	22.69	82.38
0.85	22.65	82.38
0.9	22.61	82.38
0.95	22.58	82.38
1	22.55	82.37



L [m]	$T_{w,out}$ [C]	$T_{air,out}$ [C]
5	24.35	86.76
6	24.8	86.14
7	25.24	85.53
8	25.67	84.93
9	26.1	84.35
10	26.52	83.77
11	26.93	83.2
12	27.34	82.64
13	27.74	82.09
14	28.13	81.54
15	28.52	81.01
16	28.9	80.48
17	29.28	79.96
18	29.65	79.45
19	30.01	78.95
20	30.37	78.45
21	30.73	77.96
22	31.08	77.48
23	31.42	77
24	31.76	76.53
25	32.1	76.07



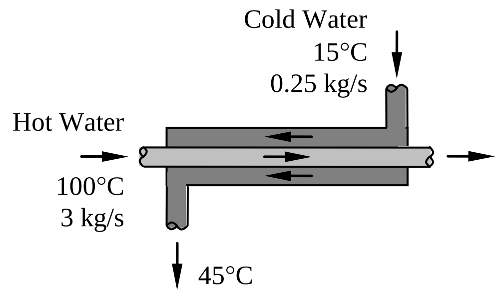
11-99 Cold water is heated by hot water in a heat exchanger. The net rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

Properties The specific heats of the cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_c = \dot{m}_c c_{pc} = (0.25\ \text{kg/s})(4180\ \text{J/kg.°C}) = 1045\ \text{W/°C}$$
$$C_h = \dot{m}_h c_{ph} = (3\ \text{kg/s})(4190\ \text{J/kg.°C}) = 12{,}570\ \text{W/°C}$$



Therefore,

$$C_{\min} = C_c = 1045\ \text{W/°C}$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{1045}{12{,}570} = 0.083$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (1045\ \text{W/°C})(100\text{°C} - 15\text{°C}) = 88{,}825\ \text{W}$$

The actual rate of heat transfer is

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) = (1045\ \text{W/°C})(45\text{°C} - 15\text{°C}) = \mathbf{31{,}350W}$$

Then the effectiveness of this heat exchanger becomes

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{31{,}350}{88{,}825} = 0.35$$

The NTU of this heat exchanger is determined using the relation in Table 11-5 to be

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.083-1}\ln\left(\frac{0.35-1}{0.35\times 0.083-1}\right) = 0.438$$

Then the surface area of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{\min}} \longrightarrow A = \frac{NTU\,C_{\min}}{U} = \frac{(0.438)(1045\ \text{W/°C})}{950\ \text{W/m}^2.\text{°C}} = \mathbf{0.482m^2}$$

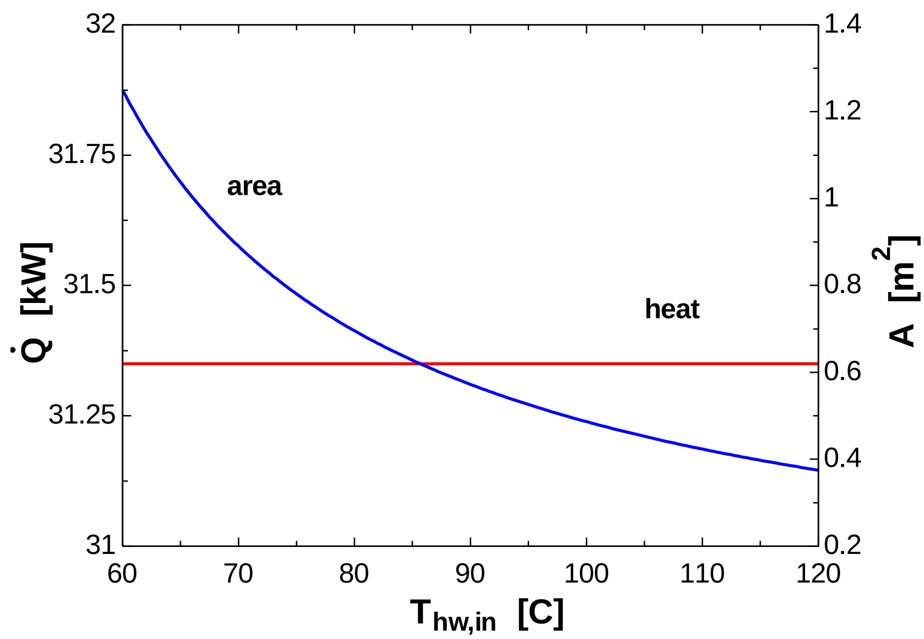
11-100 Prob. 11-99 is reconsidered. The effects of the inlet temperature of hot water and the heat transfer coefficient on the rate of heat transfer and the surface area are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

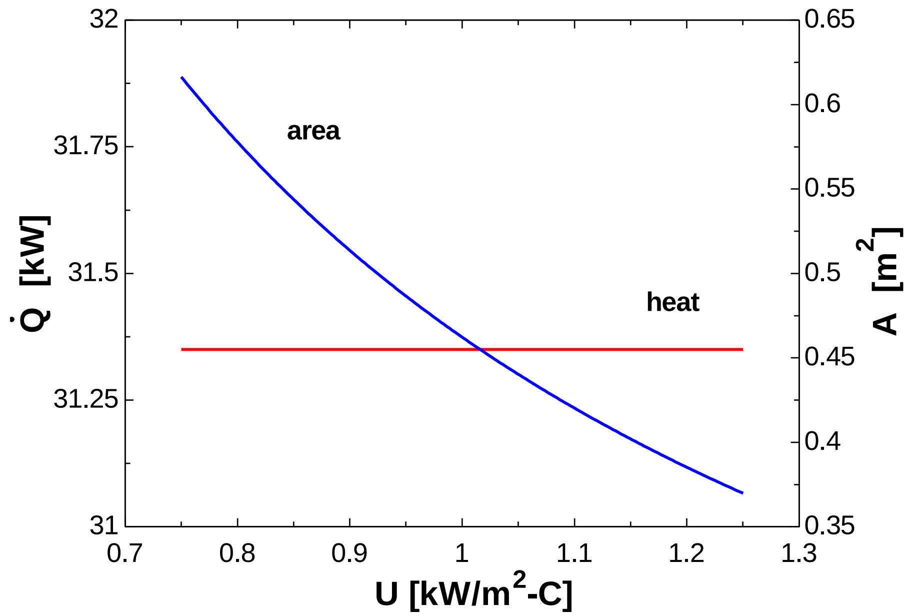
```
"GIVEN"
T_cw_in=15 [C]
T_cw_out=45 [C]
m_dot_cw=0.25 [kg/s]
c_p_cw=4.18 [kJ/kg-C]
T_hw_in=100 [C]
m_dot_hw=3 [kg/s]
c_p_hw=4.19 [kJ/kg-C]
U=0.95 [kW/m^2-C]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use LMTD method. Both methods give the same results."
DELTAT_1=T_hw_in-T_cw_out
DELTAT_2=T_hw_out-T_cw_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A*DELTAT_lm
Q_dot=m_dot_hw*c_p_hw*(T_hw_in-T_hw_out)
Q_dot=m_dot_cw*c_p_cw*(T_cw_out-T_cw_in)
```

$T_{hw, in}$ [C]	$\dot{Q}$ [kW]	A [m^2]
60	31.35	1.25
65	31.35	1.038
70	31.35	0.8903
75	31.35	0.7807
80	31.35	0.6957
85	31.35	0.6279
90	31.35	0.5723
95	31.35	0.5259
100	31.35	0.4865
105	31.35	0.4527
110	31.35	0.4234
115	31.35	0.3976
120	31.35	0.3748

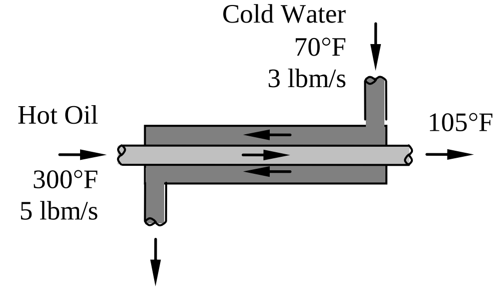


U [kW/m^2-C]	$\dot{Q}$ [kW]	A [m^2]
0.75	31.35	0.6163
0.8	31.35	0.5778
0.85	31.35	0.5438
0.9	31.35	0.5136
0.95	31.35	0.4865
1	31.35	0.4622
1.05	31.35	0.4402
1.1	31.35	0.4202
1.15	31.35	0.4019
1.2	31.35	0.3852
1.25	31.35	0.3698



11-101E Oil is cooled by water in a double-pipe heat exchanger. The overall heat transfer coefficient of this heat exchanger is to be determined using both the LMTD and NTU methods.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The thickness of the tube is negligible since it is thin-walled.



Properties The specific heats of the water and oil are given to be 1.0 and 0.525 Btu/lbm.°F, respectively.

Analysis (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in-}T_{h,out}) = (5\text{ lbm/s})(0.525\text{ Btu/lbm.°F})(300-105\text{°F}) = 511.9\text{ Btu/s}$$

The outlet temperature of the cold fluid is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 70\text{°F} + \frac{511.9\text{ Btu/s}}{(3\text{ lbm/s})(1.0\text{ Btu/lbm.°F})} = 240.6\text{°F}$$

The temperature differences between the two fluids at the two ends of the heat exchanger are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 300\text{°F} - 240.6\text{°F} = 59.4\text{°F}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 105\text{°F} - 70\text{°F} = 35\text{°F}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{59.4 - 35}{\ln(59.4/35)} = 46.1\text{°F}$$

Then the overall heat transfer coefficient becomes

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow U = \frac{\dot{Q}}{A_s \Delta T_{lm}} = \frac{511.9\text{ Btu/s}}{\pi(5/12\text{ m})(200\text{ ft})(46.1\text{ °F})} = \mathbf{0.0424 Btu/s.ft^2.°F}$$

(*b*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (5\text{ lbm/s})(0.525\text{ Btu/lbm.°F}) = 2.625\text{ Btu/s.°F}$$

$$C_c = \dot{m}_c c_{pc} = (3\text{ lbm/s})(1.0\text{ Btu/lbm.°F}) = 3.0\text{ Btu/s.°F}$$

Therefore, $C_{\min} = C_h = 2.625\text{ Btu/s.°F}$ and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{2.625}{3.0} = 0.875$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (2.625\text{ Btu/s.°F})(300\text{°F - }70\text{°F}) = 603.75\text{ Btu/s}$$

The actual rate of heat transfer and the effectiveness are

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = (2.625\text{ Btu/s.°F})(300\text{°F - }105\text{°F}) = 511.9\text{ Btu/s}$$

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{511.9}{603.75} = 0.85$$

The NTU of this heat exchanger is determined using the relation in Table 11-5 to be

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.875-1}\ln\left(\frac{0.85-1}{0.85\times 0.875-1}\right) = 4.28$$

The heat transfer surface area of the heat exchanger is

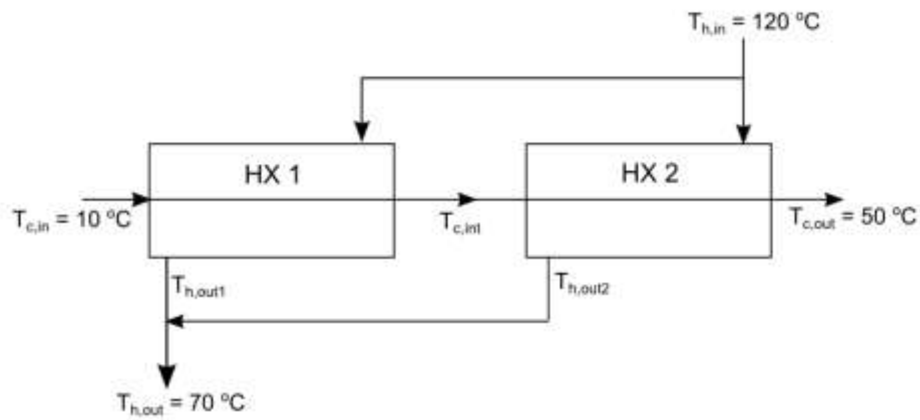
$$A_s = \pi DL = \pi(5/12\text{ ft})(200\text{ ft}) = 261.8\text{ ft}^2$$

and $$NTU = \frac{UA_s}{C_{\min}} \longrightarrow U = \frac{NTU\, C_{\min}}{A_s} = \frac{(4.28)(2.625\text{ Btu/s.°F})}{261.8\text{ ft}^2} = \mathbf{0.0429 Btu/s.ft^2.°F}$$

11-102 For the given set of flow conditions, the effectiveness, NTU and surface area of a large heat exchanger and its proposed replacement of two small heat exchangers arranged in series is to be determined. Based on the surface area and construction cost, economic choice of the heat exchanger is to be made.

Assumptions **1** Steady state conditions exist. **2** Fluid properties remain constant. **3** Heat exchanger is well insulated. **4** Negligible fouling resistance.

Properties Specific heat of glycerin is calculated at an inlet and exit average temperature of (10 + 50)°C/2 = 30°C from Table A-13. c_{pc} = 2447 J/kg·K.



Analysis Let's first consider the **large single heat exchanger**. The energy balance within the heat exchanger gives

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\therefore \dot{m}_h c_{ph} \times (120 - 70)^{\circ}\text{C} = (1.5\text{kg/s})(2447\ \text{J/kg}\cdot\text{K})\,(50 - 10)^{\circ}\text{C} = 146.82\ \text{kW}$$

Thus the heat capacity rate of the hot fluid is

$$C_h = \dot{m}_h c_{ph} = 2936.4\ \text{W/K}$$

And that for the cold fluid is

$$C_c = \dot{m}_c c_{pc} = (1.5\text{kg/s})\,(2447\ \text{J/kg}\cdot\text{K}) = 3670.5\ \ \text{W/K}.$$

Thus

$$C_{min} = C_h.$$

Thus the effectiveness of the large single heat exchanger is,

(a) $$\varepsilon = \frac{\dot{Q}}{Q_{\max}} = \frac{C_c\,(T_{c,out} - T_{c,in})}{C_{\min}\,(T_{h,in} - T_{c,in})} = \frac{(3670.5\text{W/K})(50-10)^{\circ}\text{C}}{(2936.4\text{W/K})(120-10)^{\circ}\text{C}} = \mathbf{0.455}$$

The capacity ratio of the large single heat exchanger is,

$$c = \frac{C_{\min}}{C_{\max}} = \frac{2936.4\,\text{W/K}}{3670.5\,\text{W/K}} = 0.8$$

The number of transfer units (NTU) for single heat exchanger is determined from the counter flow relation in Table 11-5.

(b) $$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon\,c - 1}\right) = \frac{1}{0.8-1}\ln\left(\frac{0.455-1}{0.455\times 0.8 - 1}\right) = \mathbf{0.77}$$

The heat transfer is area is now calculated as,

(c) $$A_s = \frac{NTU\,C_{\min}}{U} = \frac{(0.77)(2936.4\,\text{W/K})}{950\,\text{W/m}^2\cdot\text{K}} = \mathbf{2.38}\ \text{m}^2$$

Now let's consider the **two small heat exchangers** connected in series.

Given that the flow of water is split between the two heat exchangers such that 60% would go to the first heat exchanger and the remaining 40% would go to the second heat exchanger. The heat capacity rate of the 1st heat exchanger is C_{h1} = 0.6 (2936.4) = 1761.8 W/K while that of the 2nd heat exchanger is C_{h2} = 0.4 (2936.4) = 1174.6 W/K . The heat capacity rate of the cold side remains unchanged i.e. 3670.5 W/K.

The capacity ratio of the 1st heat exchangers is

$$c_1 = \frac{C_{\min}}{C_{\max}} = \frac{1761.8}{3670.5} = 0.48$$

The capacity ratio of the 2nd heat exchangers is

$$c_2 = \frac{C_{\min}}{C_{\max}} = \frac{1174.6}{3670.5} = 0.32$$

The effectiveness of 1st heat exchanger is,

$$\varepsilon_1 = \frac{\dot{Q}}{Q_{\max}} = \frac{C_c(T_{c,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{(3670.5\,\text{W/K})(T_{c,\text{int}} - 10)^\text{o}\,\text{C}}{(1761.8\,\text{W/K})(120 - 10)^\text{o}\,\text{C}} = 0.0189(T_{c,\text{int}} - 10) \quad (1)$$

The effectiveness of the 2nd heat exchanger is,

$$\varepsilon_2 = \frac{\dot{Q}}{Q_{\max}} = \frac{C_c(T_{c,out} - T_{c,\text{int}})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{(3670.5\,\text{W/K})(50 - T_{c,\text{int}})^\text{o}\,\text{C}}{(1174.6\,\text{W/K})(120 - T_{c,\text{int}})^\text{o}\,\text{C}} = 3.125\frac{(50 - T_{c,\text{int}})}{(120 - T_{c,\text{int}})} \quad (2)$$

Since the total water flow rate is split into 60% in the first heat exchanger and 40% in the second heat exchanger, the average exit temperature of the hot water from an energy balance is,

$$\dot{m}_{h,out1}\, c_{ph}\; T_{h,out1} + \dot{m}_{h,out1} c_{ph}\; T_{h,out1} = \dot{m}_{h,out} c_{ph}\; T_{h,out}$$

where c_{ph} is constant evaluated at the average inlet and outlet temperature of the hot stream

$$0.6\dot{m}_{h,out} c_{ph}\; T_{h,out1} + 0.4\dot{m}_{h,out}\, c_{ph}\; T_{h,out1} = \dot{m}_{h,out} c_{ph}\; T_{h,out}$$

$$\therefore\quad 0.6T_{h,out1} + 0.4T_{h,out2} = 70^o C$$

Further, the energy balance on the 1st heat exchanger gives

$$\dot{m}_h c_{ph,1}(T_{h,in} - T_{h,out1}) = \dot{m}_c c_{pc}(T_{c,\text{int}} - T_{c,in})$$
$$\therefore (1761.8\,\text{W/K})(120 - T_{h,out1})^\text{o}\,\text{C} = (3670.5\,\text{W/K})(T_{c,\text{int}} - 10)^\text{o}\,\text{C} \quad (3)$$

For 2nd heat exchanger, the energy balance is,

$$\dot{m}_h C_{ph,1}(T_{h,in} - T_{h,out2}) = \dot{m}_c C_{pc}(T_{c,out} - T_{c,\text{int}})$$
$$\therefore (1174.6\,\text{W/K})(120 - T_{h,out2}) = (3670.5\,\text{W/K})(50 - T_{c,\text{int}}) \quad (4)$$

The number of transfer units for 1st heat exchanger is

$$NTU_1 = \frac{1}{c_1 - 1}\ln\left(\frac{\varepsilon_1 - 1}{c_1\varepsilon_1 - 1}\right) \quad (5)$$

The number of transfer units for 2nd heat exchanger is

$$NTU_2 = \frac{1}{c_2 - 1}\ln\left(\frac{\varepsilon_2 - 1}{c_2\varepsilon_2 - 1}\right) \quad (6)$$

Further, since the surface area of each heat exchanger is same i.e., $A_{s1} = A_{s2}$ we get,

$$\frac{NTU_1 C_{ph,1}}{U} = \frac{NTU_2 C_{ph,2}}{U} \quad (7)$$

Solving Equations (1) to (7) simultaneously in EES or any other software the exit temperature of hot fluid in each heat exchanger and the intermediate temperature of the cold fluid is,

$$T_{h,out1} = 70.82\ ^oC, T_{h,out2} = 68.77\ ^oC, T_{c,\text{int}} = 33.61\ ^oC$$

(a) The effectiveness of 1st and 2nd heat exchanger is **0.447** and **0.593**, respectively.

(b) The number of transfer units (NTU) of 1st and 2nd heat exchanger is **0.675** and **1.012**, respectively.

(c) The surface area of each small heat exchanger is **1.252 m^2**.

(d) The heat exchanger surface area of 1.252 m^2 is for one heat exchanger. Thus for two heat exchangers arranged in series the total surface area is 2.5 m^2. This area is about 5% higher than that of a single large heat exchanger. Moreover, the construction cost of a small heat exchanger is about 15% higher than a large heat exchanger per unit surface area. This translates to about 2% increase in the construction cost. Hence, it is recommended to use one large heat exchanger instead of two heat exchangers arranged in series.

11-103E A 1-shell and 2-tube type heat exchanger has a specified overall heat transfer coefficient, (*a*) the heat transfer effectiveness and (*b*) the actual heat transfer rate in the heat exchanger are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Analysis (*a*) The heat capacity rates are given as

$$C_{\min} = 20{,}000\ \text{Btu/hr}\cdot°\text{F} \qquad \text{and} \qquad C_{\max} = 40{,}000\ \text{Btu/hr}\cdot°\text{F}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{20{,}000\ \text{Btu/hr}\cdot°\text{F}}{40{,}000\ \text{Btu/hr}\cdot°\text{F}} = 0.5$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(300\ \text{Btu/hr}\cdot\text{ft}^2\cdot°\text{F})(100\ \text{ft}^2)}{20{,}000\ \text{Btu/hr}\cdot°\text{F}} = 1.5$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\varepsilon = 2\left\{1 + c + \sqrt{1+c^2}\,\frac{1+\exp[-\text{NTU}\sqrt{1+c^2}\,]}{1-\exp[-\text{NTU}\sqrt{1+c^2}\,]}\right\}^{-1}$$

$$= 2\left\{1 + 0.5 + \sqrt{1+0.5^2}\,\frac{1+\exp[-1.5\sqrt{1+0.5^2}\,]}{1-\exp[-1.5\sqrt{1+0.5^2}\,]}\right\}^{-1} = \mathbf{0.639 = 63.9\%}$$

(*b*) The maximum possible heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{h,\text{in}} - T_{c,\text{in}}) = (20{,}000\ \text{Btu/hr}\cdot°\text{F})(200-90)°\text{F} = 2.20\times10^6\ \text{Btu/hr}$$

Hence, the actual heat transfer rate in the heat exchanger is

$$\dot{Q} = \varepsilon\dot{Q}_{\max} = (0.639)(2.20\times10^6\ \text{Btu/hr}) = \mathbf{1.41\times10^6\ Btu/hr}$$

Discussion Using Figure 11-27*c*, the heat transfer effectiveness is verified to be $\varepsilon \approx 64\%$.

11-104 Cold water is being heated in a 1-shell and 2-tube heat exchanger, the outlet temperatures of the cold water and hot water are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heats of the cold water and hot water are given to be c_{pc} = 4178 J/kg·K and c_{ph} = 4188 J/kg·K, respectively.

Analysis The heat capacity rates are

$$C_c = \dot{m}_c c_{pc} = (5000\text{ kg/h})(1/3600\text{ h/s})(4178\text{ J/kg}\cdot\text{K}) = 5802.8\text{ W/K}$$

$$C_h = \dot{m}_h c_{ph} = (10{,}000\text{ kg/h})(1/3600\text{ h/s})(4188\text{ J/kg}\cdot\text{K}) = 11{,}633\text{ W/K}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{C_c}{C_h} = \frac{5802.8\text{ W/K}}{11{,}633\text{ W/K}} = 0.499$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{11{,}600\text{ W/K}}{5802.8\text{ W/K}} = 1.999$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\varepsilon = 2\left\{1 + c + \sqrt{1+c^2}\,\frac{1+\exp[-\text{NTU}\sqrt{1+c^2}\,]}{1-\exp[-\text{NTU}\sqrt{1+c^2}\,]}\right\}^{-1}$$

$$= 2\left\{1 + 0.499 + \sqrt{1+0.499^2}\,\frac{1+\exp[-1.999\sqrt{1+0.499^2}\,]}{1-\exp[-1.999\sqrt{1+0.499^2}\,]}\right\}^{-1} = 0.6933$$

The outlet temperature of the cold water is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_c(T_{h,\text{in}} - T_{c,\text{in}})} \qquad \to \qquad T_{c,\text{out}} = \varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) + T_{c,\text{in}}$$

$$T_{c,\text{out}} = \varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) + T_{c,\text{in}} = (0.6933)(80-20)°\text{C} + 20°\text{C} = \mathbf{61.9°C}$$

The outlet temperature of the hot water is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,\text{in}} - T_{h,\text{out}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_h(T_{h,\text{in}} - T_{h,\text{out}})}{C_c(T_{h,\text{in}} - T_{c,\text{in}})}$$

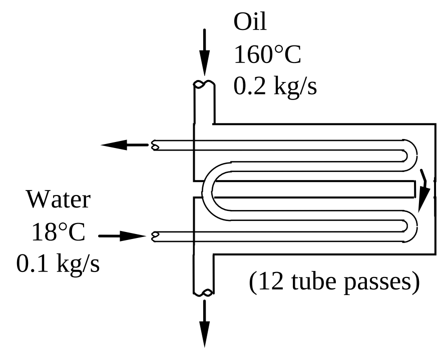
$$T_{h,\text{out}} = T_{h,\text{in}} - \frac{C_c}{C_h}\varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) = 80°\text{C} - (0.499)(0.6933)(80-20)°\text{C} = \mathbf{59.2°C}$$

Discussion Using Figure 11-27*c*, the heat transfer effectiveness is approximately $\varepsilon \approx 69\%$.

11-105 Hot oil is to be cooled by water in a heat exchanger. The mass flow rates and the inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined. √

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The thickness of the tube is negligible since it is thin-walled. **5** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.



Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.2\,\text{kg/s})(2200\ \text{J/kg.°C}) = 440\ \text{W/°C}$$
$$C_c = \dot{m}_c c_{pc} = (0.1\,\text{kg/s})(4180\ \text{J/kg.°C}) = 418\ \text{W/°C}$$

Therefore,

$$C_{\min} = C_c = 418\ \text{W/°C}$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{418}{440} = 0.95$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (418\ \text{W/°C})(160\text{°C} - 18\text{°C}) = 59.36\,\text{kW}$$

The heat transfer surface area is

$$A_s = n(\pi DL) = (12)(\pi)(0.018\,\text{m})(3\,\text{m}) = 2.04\,\text{m}^2$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(340\ \text{W/m}^2.\text{°C})\,(2.04\,\text{m}^2)}{418\ \text{W/°C}} = 1.659$$

Then the effectiveness of this heat exchanger corresponding to c = 0.95 and NTU = 1.659 is determined from Fig. 11-27d to be

$$\varepsilon = 0.61$$

Then the actual rate of heat transfer becomes

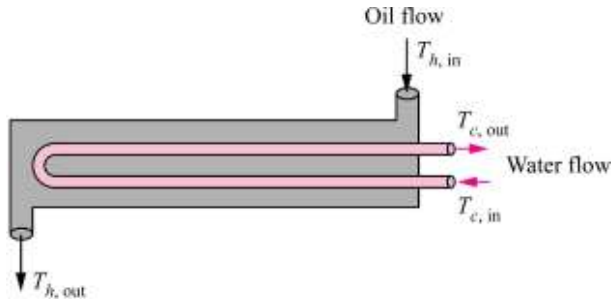
$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.61)(59.36\,\text{kW}) = \mathbf{36.2\,kW}$$

Finally, the outlet temperatures of the cold and hot fluid streams are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 18\text{°C} + \frac{36.2\,\text{kW}}{0.418\,\text{kW / °C}} = \mathbf{104.6°C}$$
$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 160\text{°C} - \frac{36.2\,\text{kW}}{0.44\,\text{kW/°C}} = \mathbf{77.7°C}$$

11-106E A 1-shell and 2-tube heat exchanger has specified overall heat transfer coefficient, inlet and outlet temperatures, and mass flow rates, (*a*) the NTU value and (*b*) the surface area of the heat exchanger are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.



Properties The specific heat of water is given to be c_{pc} = 1.0 Btu/lbm·°F.

Analysis (*a*) The heat capacity rate for the cold fluid (water) is

$$C_c = \dot{m}_c c_{pc} = (20{,}000\ \text{lbm/hr})(1.0\ \text{Btu/lbm}\cdot°\text{F}) = 20{,}000\ \text{Btu/hr}\cdot°\text{F}$$

Using energy balance, we have

$$C_c(T_{c,\text{out}} - T_{c,\text{in}}) = C_h(T_{h,\text{in}} - T_{h,\text{out}}) \qquad \rightarrow \qquad \frac{C_c}{C_h} = \frac{T_{h,\text{in}} - T_{h,\text{out}}}{T_{c,\text{out}} - T_{c,\text{in}}} = \frac{180-120}{100-80} = 3.0$$

or

$$c = \frac{C_h}{C_c} = \frac{C_{\min}}{C_{\max}} = \frac{1}{3} = 0.3333$$

The heat transfer effectiveness is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_h(T_{h,\text{in}} - T_{c,\text{in}})} = (3.0)\frac{100-80}{180-80} = 0.60$$

From Table 11-4, the NTU value can be determined from

$$\varepsilon = 2\left\{1 + c + \sqrt{1+c^2}\,\frac{1+\exp[-\text{NTU}\sqrt{1+c^2}\,]}{1-\exp[-\text{NTU}\sqrt{1+c^2}\,]}\right\}^{-1}$$

Copy the following lines and paste on a blank EES screen to solve the above equation:

```
c=1/3
epsilon=0.60
epsilon=2*(1+c+sqrt(1+c^2)*(1+exp(-NTU*sqrt(1+c^2)))/(1-exp(-NTU*sqrt(1+c^2))))^(-1)
```

Solving by EES software, we get

$$\text{NTU} = \mathbf{1.11}$$

(*b*) The surface area of the heat exchanger can be determined using

$$\text{NTU} = \frac{UA_s}{C_{\min}} \qquad \rightarrow \qquad A_s = \text{NTU}\frac{C_{\min}}{U} = \text{NTU}\frac{cC_c}{U}$$

$$A_s = \text{NTU}\frac{cC_c}{U} = (1.11)\frac{(1/3)(20{,}000\ \text{Btu/hr}\cdot°\text{F})}{40\ \text{Btu/hr}\cdot\text{ft}^2\cdot°\text{F}} = \mathbf{185\ ft^2}$$

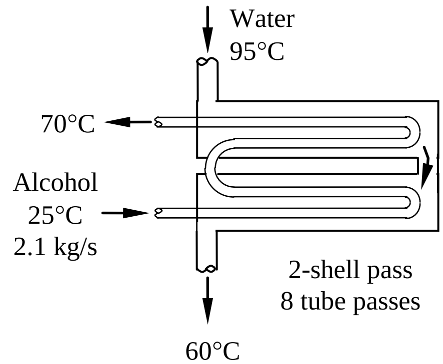
Discussion Using Figure 11-27*c*, the NTU value is found to be approximately NTU ≈ 1.2.

11-107 Ethyl alcohol is heated by water in a shell-and-tube heat exchanger. The heat transfer surface area of the heat exchanger is to be determined using both the LMTD and NTU methods.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the ethyl alcohol and water are given to be 2.67 and 4.19 kJ/kg.°C, respectively.

Analysis (*a*) The temperature differences between the two fluids at the two ends of the heat exchanger are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 95°\text{C} - 70°\text{C} = 25°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 25°\text{C} = 35°\text{C}$$

The logarithmic mean temperature difference and the correction factor are

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{25-35}{\ln(25/35)} = 29.7°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{70-25}{95-25} = 0.64 \\ R &= \frac{T_2 - T_1}{t_1 - t_1} = \frac{95-60}{70-25} = 0.78 \end{aligned}\right\} F = 0.93$$

The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (2.1\,\text{kg/s})(2.67\,\text{kJ/kg.°C})(70°\text{C} - 25°\text{C}) = 252.3\,\text{kW}$$

The surface area of heat transfer is

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm}} = \frac{252.3\,\text{kW}}{0.8\,\text{kW/m}^2.°\text{C})(0.93)(29.7°\text{C})} = \mathbf{11.4\,m^2}$$

(*b*) The rate of heat transfer is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (2.1\,\text{kg/s})(2.67\,\text{kJ/kg.°C})(70°\text{C} - 25°\text{C}) = 252.3\,\text{kW}$$

The mass flow rate of the hot fluid is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) \rightarrow \dot{m}_h = \frac{\dot{Q}}{c_{ph}(T_{h,in} - T_{h,out})} = \frac{252.3\,\text{kW}}{(4.19\,\text{kJ/kg.°C})(95°\text{C} - 60°\text{C})} = 1.72\,\text{kg/s}$$

The heat capacity rates of the hot and the cold fluids are

$$C_h = \dot{m}_h c_{ph} = (1.72\,\text{kg/s})(4.19\,\text{kJ/kg.°C}) = 7.21\,\text{kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (2.1\,\text{kg/s})(2.67\,\text{kJ/kg.°C}) = 5.61\,\text{kW/°C}$$

Therefore, $C_{\min} = C_c = 5.61\,\text{W/°C}$ and $c = \dfrac{C_{\min}}{C_{\max}} = \dfrac{5.61}{7.21} = 0.78$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (5.61\,\text{W/°C})(95°\text{C} - 25°\text{C}) = 392.7\,\text{kW}$$

The effectiveness of this heat exchanger is $\varepsilon = \dfrac{Q}{Q_{\max}} = \dfrac{252.3}{392.7} = 0.64$

The NTU of this heat exchanger corresponding to this emissivity and c = 0.78 is determined from 11-27d to be NTU = 1.7. Then the surface area of heat exchanger is determined to be

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{NTU\,C_{\min}}{U} = \frac{(1.7)(5.61\,\text{kW/°C})}{0.8\,\text{kW/m}^2.°\text{C}} = \mathbf{11.9\,m^2}$$

The small difference between the two results is due to the reading error of the chart.

11-108 Cold water is heated by hot oil in a shell-and-tube heat exchanger. The rate of heat transfer is to be determined using both the LMTD and NTU methods.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.

Analysis (*a*) The LMTD method in this case involves iterations, which involves the following steps:

1) Choose $T_{h,out}$

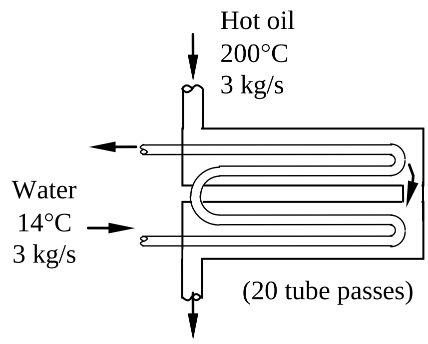
2) Calculate $\dot{Q}$ from $\dot{Q} = \dot{m}_h c_p (T_{h,out} - T_{h,in})$

3) Calculate $T_{h,out}$ from $\dot{Q} = \dot{m}_h c_p (T_{h,out} - T_{h,in})$

4) Calculate $\Delta T_{lm,CF}$

5) Calculate $\dot{Q}$ from $\dot{Q} = UA_s F \Delta T_{lm,CF}$

6) Compare to the $\dot{Q}$ calculated at step 2, and repeat until reaching the same result



Result: **651 kW**

(*b*) The heat capacity rates of the hot and the cold fluids are

$$C_h = \dot{m}_h c_{ph} = (3 \text{ kg/s})(2.2 \text{ kJ/kg.°C}) = 6.6 \text{ kW/°C}$$
$$C_c = \dot{m}_c c_{pc} = (3 \text{ kg/s})(4.18 \text{ kJ/kg.°C}) = 12.54 \text{ kW/°C}$$

Therefore,

$$C_{\min} = C_h = 6.6 \text{ kW/°C}$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{6.6}{12.54} = 0.53$$

Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (6.6 \text{ kW/°C})(200\text{°C} - 14\text{°C}) = 1228 \text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.3 \text{ kW/m}^2\text{.°C})(20 \text{ m}^2)}{6.6 \text{ kW/°C}} = 0.91$$

Then the effectiveness of this heat exchanger corresponding to $c = 0.53$ and NTU = 0.91 is determined from Fig. 11-27d to be

$$\varepsilon = 0.53$$

The actual rate of heat transfer then becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.53)(1228 \text{ kW}) = \mathbf{651 kW}$$

11-109E In a one shell pass and eight tube pass heat exchanger, water is to be heated using hot air at 600°F. For the given value of convection heat transfer coefficient on the outer surface of the tubes and the fouling resistances, the heat exchanger area is to be determined using LMTD and $\varepsilon - NTU$ methods.

Assumptions **1** Steady state conditions exist. **2** Fluid properties remain constant. **3** Heat exchanger is well insulated.

Properties Calculate the thermal properties of water at an average temperature of (150 + 70)°F/2 = 110°F from Table 9-E. c_{pc} = 0.999 Btu/1bm·°F ≈ 1 Btu/1bm·°F

Analysis The heat gained by water from hot air is,

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (50{,}000\,\text{lbm/h})(1\,\text{Btu/lbm}\cdot{}^\circ F)(150 - 70)^\circ\text{F} = 4\times 10^6\ \text{Btu/h}$$

From energy balance we have, heat lost by air = heat gained by water.

$$\dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\dot{m}_h = \frac{\dot{m}_c c_{pc}(T_{c,out} - T_{c,in})}{c_{ph}(T_{h,in} - T_{h,out})} = \frac{4\times 10^6\,\text{Btu/h}}{(0.25\,\text{Btu/lbm}\cdot{}^\circ\text{F})(600 - 300)^\circ\text{F}} = 53{,}333.33\text{lbm/h}$$

Now the logarithmic temperature difference for a counter flow heat exchanger is calculated as

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 600 - 150 = 450\ {}^\text{o}\text{F}$$

and

$$\Delta T_2 = T_{h,out} - T_{c,in} = 300 - 70 = 230^\text{o}\,\text{F}$$

$$\Delta T_{lm,CF} = \frac{450 - 230}{\ln(450/230)} = 327.78\ {}^\text{o}\text{F}$$

In order to determine correction factor we first need to find the temperature ratios P and R as follows:

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{300 - 600}{70 - 600} = 0.566 \quad \text{and} \quad R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{70 - 150}{300 - 600} = 0.266$$

From Figure 11-19 (a), the correction factor is

F = 0.96.

The heat transfer rate is

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$

The overall heat transfer coefficient is to be calculated from the given value of convection heat transfer coefficient on the outer surface of the tubes and accounting for the possible fouling resistance on both water and air side.

$$\frac{1}{U} = \frac{1}{h} + R_{f,water} + R_{f,air}$$

$$\therefore \frac{1}{U} = \frac{1}{30\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}} + (0.0015\,\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu}) + (0.001\,\text{h}\cdot\text{ft}^2\cdot{}^\circ\text{F/Btu}) \rightarrow \text{U} = 27.9\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}$$

Therefore, the surface area of the heat exchanger is,

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm,CF}} = \frac{4\times 10^6\ \text{Btu/h}}{(27.9\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})(0.96)(327.78^\text{o}\,\text{F})} = \mathbf{455.5\,ft^2}$$

To use $\varepsilon - NTU$ method we first need to calculate the capacity ratio and effectiveness.

The heat capacity rate of the cold fluid is

$$C_c = \dot{m}_c c_{pc} = (50{,}000\ \text{lbm/h})(1\ \text{Btu/lbm}\cdot{}^\circ\text{F}) = 50{,}000\ \text{Btu/h}\cdot{}^\circ\text{F}$$

The heat capacity rate of the hot fluid is

$$C_h = \dot{m}_h c_{ph} = (53333.33\ \text{lbm/h})(0.25\ \text{Btu/lbm}\cdot{}^\circ\text{F}) = 13333.33\ \text{Btu/h}\cdot{}^\circ\text{F}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{13333.33}{50{,}000} = 0.266$$

The effectiveness of the heat exchanger is,

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h (T_{h,in} - T_{h,out})}{C_{\min} (T_{h,in} - T_{c,in})} = \frac{(600-300)}{(600-70)} = 0.566$$

The NTU of the heat exchanger is calculated using expressions given in Table 11-5.

$$NTU = -\frac{1}{\sqrt{1+c^2}} \ln\left(\frac{2/\varepsilon - 1 - c - \sqrt{1+c^2}}{2/\varepsilon - 1 - c + \sqrt{1+c^2}}\right)$$

$$\therefore NTU = -\frac{1}{\sqrt{1+0.266^2}} \ln\left(\frac{2/0.566 - 1 - 0.266 - \sqrt{1+0.266^2}}{2/0.566 - 1 - 0.266 + \sqrt{1+0.266^2}}\right) = -\frac{1}{1.034} \ln\left(\frac{1.233}{3.301}\right) = 0.952$$

From definition of NTU we get,

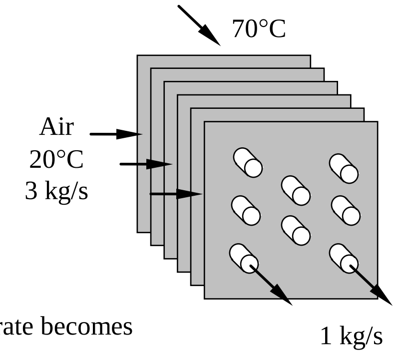
$$NTU = \frac{UA_s}{C_{\min}} \rightarrow A_s = \frac{NTU \times C_{\min}}{U} = \frac{(0.952)\ (13333.33\ \text{Btu/h}\cdot{}^\circ\text{F})}{27.9\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})} = \mathbf{454.95\ ft^2}$$

Discussion The surface area of the heat exchanger calculated using LMTD and NTU methods is within 0.1% of each other.

11-110 Air is heated by a hot water stream in a cross-flow heat exchanger. The maximum heat transfer rate and the outlet temperatures of the cold and hot fluid streams are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and air are given to be 4.19 and 1.005 kJ/kg.°C.



Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (1\,\text{kg/s})(4190\,\text{J/kg.°C}) = 4190\,\text{W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (3\,\text{kg/s})(1005\,\text{J/kg.°C}) = 3015\,\text{W/°C}$$

Therefore

$$C_{\min} = C_c = 3015\,\text{W/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (3015\,\text{W/°C})(70°\text{C} - 20°\text{C}) = 150{,}750\,\text{W} = \mathbf{150.8\,kW}$$

The outlet temperatures of the cold and the hot streams in this limiting case are determined to be

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20°\text{C} + \frac{150.75\,\text{kW}}{3.015\,\text{kW/°C}} = \mathbf{70°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 70°\text{C} - \frac{150.75\,\text{kW}}{4.19\,\text{kW/°C}} = \mathbf{34.0°C}$$

11-111 A cross-flow heat exchanger with both fluids unmixed has a specified overall heat transfer coefficient, and the exit temperature of the cold fluid is to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Analysis The heat capacity rates are given as

$$C_h = C_{\min} = 40{,}000 \text{ W/K} \quad \text{and} \quad C_c = C_{\max} = 80{,}000 \text{ W/K}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{C_h}{C_c} = \frac{40{,}000 \text{ W/K}}{80{,}000 \text{ W/K}} = 0.5$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(200 \text{ W/m}^2\cdot\text{K})(400 \text{ m}^2)}{40{,}000 \text{ W/K}} = 2.0$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\varepsilon = 1 - \exp\left\{\frac{\text{NTU}^{0.22}}{c}[\exp(-c\,\text{NTU}^{0.78}) - 1]\right\}$$

$$= 1 - \exp\left(\frac{2.0^{0.22}}{0.5}\{\exp[-(0.5)(2.0)^{0.78}] - 1\}\right) = 0.7388$$

From the definition of heat transfer effectiveness,

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_h(T_{h,\text{in}} - T_{c,\text{in}})}$$

or

$$T_{c,\text{out}} = \frac{C_h}{C_c}\varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) + T_{c,\text{in}} = (0.5)(0.7388)(80 - 20)°\text{C} + 20°\text{C} = \mathbf{42.2°C}$$

Discussion Using Figure 11-27*e*, the heat transfer effectiveness is approximately $\varepsilon \approx 73\%$.

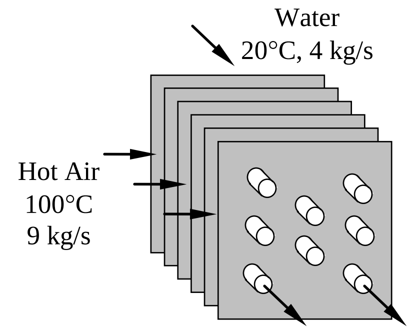
11-112 Water is heated by hot air in a heat exchanger. The mass flow rates and the inlet temperatures are given. The heat transfer surface area of the heat exchanger on the water side is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and air are given to be 4.18 and 1.01kJ/kg.°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (4\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 16.72\text{ kW/°C}$$
$$C_c = \dot{m}_c c_{pc} = (9\text{ kg/s})(1.01\text{ kJ/kg.°C}) = 9.09\text{ kW/°C}$$



Therefore,

$$C_{\min} = C_c = 9.09\text{ kW/°C}$$

and

$$C = \frac{C_{\min}}{C_{\max}} = \frac{9.09}{16.72} = 0.544$$

Then the NTU of this heat exchanger corresponding to $c = 0.544$ and $\varepsilon = 0.65$ is determined from Fig. 11-27 to be

NTU = 1.5

Then the surface area of this heat exchanger becomes

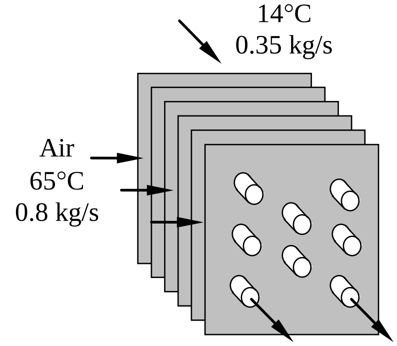
$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{\text{NTU}\, C_{\min}}{U} = \frac{(1.5)(9.09\text{ kW/°C})}{0.260\text{ kW/m}^2\text{.°C}} = \mathbf{52.4\ m^2}$$

11-113 Water is heated by a hot water stream in a heat exchanger. The maximum outlet temperature of the cold water and the effectiveness of the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and air are given to be 4.18 and 1.0 kJ/kg.°C.

Analysis The heat capacity rates of the hot and cold fluids are



$$C_h = \dot{m}_h c_{ph} = (0.8\,\text{kg/s})(1.0\,\text{kJ/kg.°C}) = 0.8\,\text{kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.35\,\text{kg/s})(4.18\,\text{kJ/kg.°C}) = 1.463\,\text{kW/°C}$$

Therefore

$$C_{\min} = C_h = 0.8\,\text{kW/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (0.8\,\text{kW/°C})(65\text{°C} - 14\text{°C}) = 40.80\,\text{kW}$$

The maximum outlet temperature of the cold fluid is determined to be

$$\dot{Q}_{\max} = C_c(T_{c,out,\max} - T_{c,in}) \longrightarrow T_{c,out,\max} = T_{c,in} + \frac{\dot{Q}_{\max}}{C_c} = 14\text{°C} + \frac{40.80\,\text{kW}}{1.463\,\text{kW/°C}} = \mathbf{41.9°C}$$

The actual rate of heat transfer and the effectiveness of the heat exchanger are

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = (0.8\,\text{kW/°C})(65\text{°C} - 25\text{°C}) = 32\,\text{kW}$$

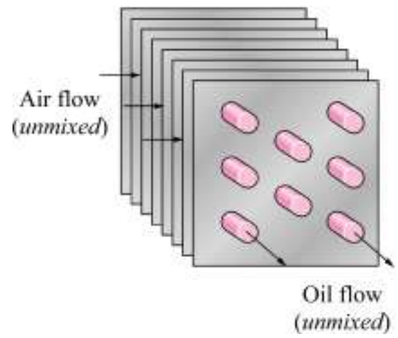
$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{32\,\text{kW}}{40.8\,\text{kW}} = \mathbf{0.784}$$

11-114 Oil in an engine is being cooled by air in a cross-flow heat exchanger, where both fluids are unmixed; (*a*) the heat transfer effectiveness and (*b*) the outlet temperature of the oil are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heats of oil and air are given to be c_{ph} = 2047 J/kg·K and c_{pc} = 1007 J/kg·K, respectively.

Analysis (*a*) The heat capacity rates are



$$C_c = \dot{m}_c c_{pc} = (0.21\,\text{kg/s})(1007\,\text{J/kg}\cdot\text{K}) = 211.5\,\text{W/K}$$

$$C_h = \dot{m}_h c_{ph} = (0.026\,\text{kg/s})(2047\,\text{J/kg}\cdot\text{K}) = 53.22\,\text{W/K}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{C_h}{C_c} = \frac{53.22\,\text{W/K}}{211.5\,\text{W/K}} = 0.2516$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(53\,\text{W/m}^2\cdot\text{K})(1\,\text{m}^2)}{53.22\,\text{W/K}} = 0.9959$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\varepsilon = 1 - \exp\left\{\frac{\text{NTU}^{0.22}}{c}[\exp(-c\,\text{NTU}^{0.78}) - 1]\right\}$$

$$= 1 - \exp\left(\frac{0.9959^{0.22}}{0.2516}\{\exp[-(0.2516)(0.9959)^{0.78}] - 1\}\right) = \mathbf{0.586}$$

(*b*) The outlet temperature of the cold water can be determined using

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,\text{in}} - T_{h,\text{out}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_h(T_{h,\text{in}} - T_{h,\text{out}})}{C_h(T_{h,\text{in}} - T_{c,\text{in}})}$$

$$T_{h,\text{out}} = T_{h,\text{in}} - \varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) = 75°\text{C} - (0.586)(75 - 30)°\text{C} = \mathbf{48.6°C}$$

Discussion Using Figure 11-27*b*, the heat transfer effectiveness is approximately $\varepsilon \approx 60\%$.

11-115 Water is heated by hot air in a cross-flow heat exchanger. Mass flow rates and inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

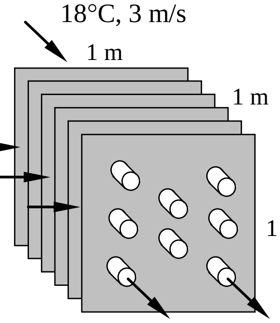
Properties The specific heats of the water and air are given to be 4.18 and 1.01 kJ/kg.°C, respectively.

Analysis The mass flow rates of the hot and the cold fluids are

$$\dot{m}_c = \rho VA_c = (1000\text{ kg/m}^3)(3\text{ m/s})[80\pi(0.03\text{ m})^2/4] = 169.6\text{ kg/s}$$

$$\rho_{air} = \frac{P}{RT} = \frac{105\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})\times(130+273\text{ K})} = 0.908\text{ kg/m}^3$$

$$\dot{m}_h = \rho VA_c = (0.908\text{ kg/m}^3)(12\text{ m/s})(1\text{ m})^2 = 10.90\text{ kg/s}$$



The heat transfer surface area and the heat capacity rates are

$$A_s = n\pi DL = 80\pi(0.03\text{ m})(1\text{ m}) = 7.540\text{ m}^2$$

$$C_c = \dot{m}_c c_{pc} = (169.6\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 708.9\text{ kW/°C}$$

$$C_h = \dot{m}_h c_{ph} = (10.9\text{ kg/s})(1.010\text{ kJ/kg.°C}) = 11.01\text{ kW/°C}$$

Therefore,

$$C_{\min} = C_c = 11.01\text{ kW/°C}$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{11.01}{708.9} = 0.01553$$

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (11.01\text{ kW/°C})(130\text{°C} - 18\text{°C}) = 1233\text{ kW}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(130\text{ W/m}^2.\text{°C})(7.540\text{ m}^2)}{11{,}010\text{ W/°C}} = 0.08903$$

Noting that this heat exchanger involves mixed cross-flow, the fluid with $C_{\min}$ is mixed, $C_{\max}$ unmixed, effectiveness of this heat exchanger corresponding to c = 0.01553 and NTU =0.08903 is determined using the proper relation in Table 11-4 to be

$$\varepsilon = 1 - \exp\left[-\frac{1}{c}(1 - e^{-cNTU})\right] = 1 - \exp\left[-\frac{1}{0.01553}(1 - e^{-0.01553\times 0.08903})\right] = 0.08513$$

Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon\dot{Q}_{\max} = (0.08513)(1233\text{ kW}) = \mathbf{105.0 kW}$$

Finally, the outlet temperatures of the cold and the hot fluid streams are determined from

$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 18\text{°C} + \frac{105.0\text{ kW}}{708.9\text{ kW / °C}} = \mathbf{18.15°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 130\text{°C} - \frac{105.0\text{ kW}}{11.01\text{ kW/°C}} = \mathbf{120.5°C}$$

11-116 Exhaust gases are used in a recuperative cross flow heat exchanger to heat air. For the given values of flow condition, heat transfer coefficients and the fouling resistance, the air exit temperature and the area of heat exchanger is to be determined.

Assumptions **1** Steady state conditions exist. **2** Heat exchanger is well insulated. **3** Fluid properties are constant.

Analysis (a) For the given flow conditions the energy balance between the two fluid streams gives,

Heat lost by exhaust gas = Heat gained by air

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\therefore T_{c,out} = \frac{\dot{m}_h c_{ph}(T_{h,in} - T_{h,out})}{\dot{m}_c c_{pc}} + T_{c,in} = \frac{(7.5\,\text{kg/s})(1069(\text{J/kg}\cdot\text{K})\,(500 - 320)°\text{C}}{(15\,\text{kg/s})(1069\,\text{J/kg}\cdot\text{K})} + 30°C = \mathbf{120^o\,C} < \mathbf{150°C}$$

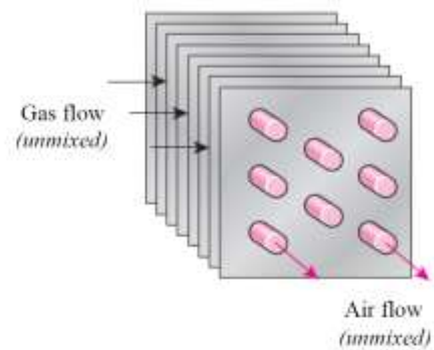
Hence the air cannot be heated to the desired temperature of 150°C.

(b) The overall heat transfer coefficient is calculated as,

$$\frac{1}{U} = \frac{1}{h_i} + R_{f,i} + \frac{1}{h_o} + R_{f,o} = \frac{1}{750\,\text{W/m}^2\cdot\text{K}} + (0.0004\,\text{m}^2\cdot\text{K/W}) + \frac{1}{300\,\text{W/m}^2\cdot\text{K}} + (0.0004\,\text{m}^2\cdot\text{K/W})$$

Therefore, the overall heat transfer coefficient is

U = 182.92 W/m^2·K.



The heat capacity rate of the cold fluid (air) is

$$C_c = \dot{m}_c c_{pc} = (15\,\text{kg/s})(1069\,\text{J/kg}\cdot\text{K}) = 16035\,\text{W/K} \rightarrow C_{\text{max}}$$

The heat capacity rate of the hot fluid (exhaust gas) is,

$$C_h = \dot{m}_h c_{ph} = (7.5\,\text{kg/s})\,(1069\,\text{J/kg}\cdot\text{K}) = 8017.5\,\text{W/K} \rightarrow C_{\text{min}}$$

Thus the capacity ratio is

$$c = \frac{C_{\text{min}}}{C_{\text{max}}} = \frac{8017.5}{16035} = 0.5$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\text{min}}(T_{h,in} - T_{c,in})} = \frac{(500-320)^o C}{(500-30)^o C} = 0.383$$

Now the number of transfer units (NTU) for both fluids unmixed are obtained from Figure 11-27(e),

NTU = 0.6

From the definition of NTU we get,

$$A_s = \frac{NTU \times C_{\text{min}}}{U} = \frac{0.6(8017.5\,\text{W/K})}{182.9\,\text{W/m}^2\cdot\text{K}} = \mathbf{26.30\,m^2}$$

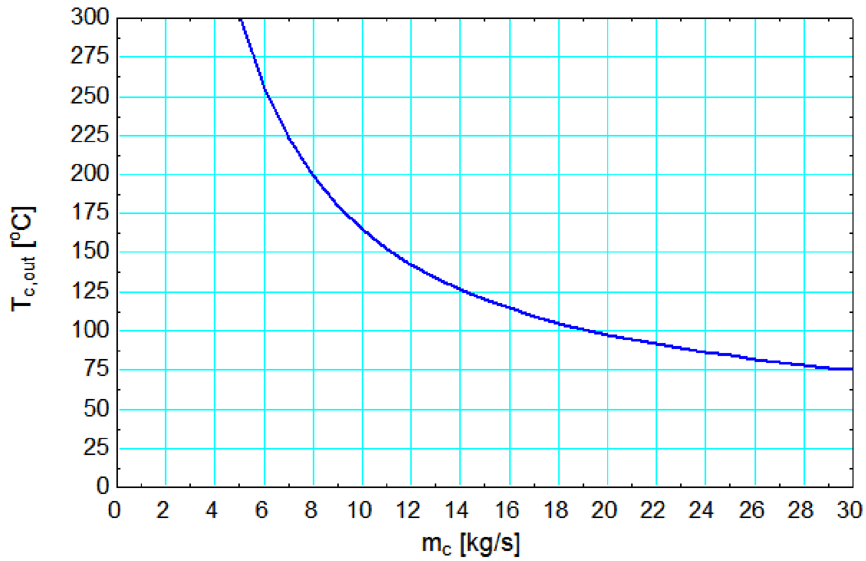
(c) The air mass flow rate in order to attain the desired exit temperature of 150°C is obtained form the energy balance of part (a).

$$\dot{m}_c = \frac{\dot{m}_h c_{ph}(T_{h,in} - T_{h,out})}{c_{pc}(T_{c,out} - T_{c,in})} = \frac{(7.5\,\text{kg/s})(1069(\text{J/kg}\cdot\text{K})\,(500 - 320)°\text{C}}{(1069\,\text{J/kg}\cdot\text{K})(T_{c,out} - 30°C)} \qquad (1)$$

For the desired $T_{\text{c,out}}$ = 150°C from the above equation we get

$\dot{m}_c$ = 11.25 kg/s

(d) Plotting equation 1 for the given range of the exit air temperature results in the following figure.



Change in air exit temperature with change in the air mass flow rate.

Discussion The plot shows that the air exit temperature is very sensitive to low air mass flow rates. It is also found that the effectiveness of the heat exchanger does not change with change in the air mass flow rate and hence increase in the air exit temperature. Since, for a cross flow heat exchanger with both fluids unmixed, the NTU is a function of effectiveness only. This implies that the NTU and hence the heat exchanger area will remain unchanged even with decrease in the air mass flow rate. However, this conclusion is in limit for $C_{min} = C_h$.

11-117 Single pass cross flow heat exchanger uses water (mixed) to heat methanol (unmixed) initially at 10°C. The heat exchanger area corresponding to the effectiveness of 0.5 is to be determined. Further, the effect of mass flow rate of water on the heat transfer rate, methanol exit temperature, overall heat transfer coefficient and the effectiveness of the heat exchanger is to be investigated.

Assumptions **1** Steady state conditions exist. **2** Heat exchanger is well insulated. **3** Fluid properties remain constan. **4** No fouling within the heat exchanger.

Analysis The effectiveness of heat exchanger is given to be 0.5.

$$\therefore 0.5 = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})}$$

Given that the total heat transfer rate is 250 kW, we find the maximum possible heat transfer as,

$$\dot{Q}_{\max} = 250\,kW / 0.5 = 500\text{ kW}$$

For the known values of $T_{h,in}$, $T_{c,in}$ and $\dot{Q}_{max}$, we get

$$C_{min} = (500 \times 10^3\text{ W})/\ (90 - 10)°\text{C} = 6250\text{ W/K.}$$

Assuming the methanol to have the minimum heat capacity rate,

$$C_c = C_{min} = 6250\text{ W/K.}$$

Now, the mass flow rate of methanol is

$$\dot{m}_c = C_c / c_{pc} = (6250\text{ W/K})/(2577\text{J/kg}\cdot\text{K}) = 2.425\text{ kg/s}$$

The mass flow rate of water can be calculated from the energy balance between water and methanol.

Heat lost by water = Heat gained by methanol

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$250 \times 10^3\text{ W} = (2.425\text{kg/s})(2577\text{ J/kg}\cdot\text{K}) \times (T_{c,out}\text{ - }10)^{\text{o}}\text{C}$$

$$\therefore T_{c,out} = 50\ ^{o}C$$

Thus the mass flow rate of water is,

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out})$$

$$250 \times 10^3\text{ W} = \dot{m}_h (4193\text{ J/kg}\cdot\text{K})(90\text{ - }60)^{\text{o}}\text{C}$$

$$\therefore \dot{m}_h = \mathbf{1.99\,kg/s}$$

The heat capacity of the water is

$$C_h = \dot{m}_h c_{ph} = 8333\text{ W/K} \to C_{\max}$$

Since C_h is C_{max} our assumption of $C_c = C_{min}$ is correct.

Thus the capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{6250}{8333} = 0.75$$

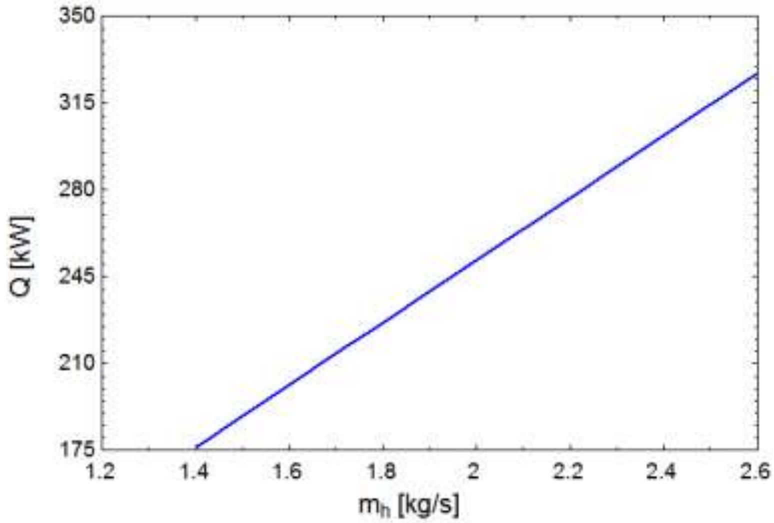
Now the number of transfer units (NTU) are calculated from the equation in Table 11-5 for C_{max} (mixed) and C_{min} (unmixed).

$$NTU = -\ln\left[1 + \frac{\ln(1 - \varepsilon\, c)}{c}\right] = -\ln\left[1 + \frac{\ln(1 - 0.5 \times 0.75)}{0.75}\right] = 0.985$$

Now from the definition of NTU we can find the surface area as

$$A_s = \frac{NTU\, C_{\min}}{U} = \frac{0.985\,(6250\text{ W/K})}{650\text{ W/m}^2\cdot\text{K}} = \mathbf{9.47\,m^2}$$

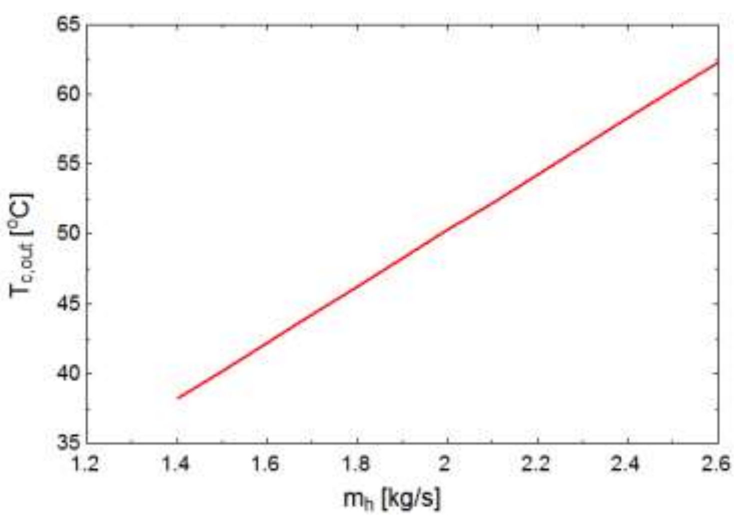
(b)



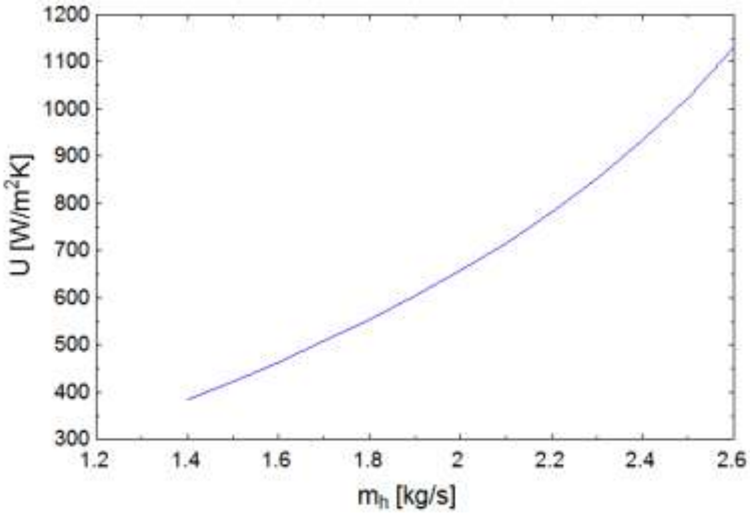
Variation of the total heat transfer rate with change in water mass flow rate

As shown in the figure above, a variation of ±30% in the water mass flow rate causes the heat transfer rate to vary by 85%. The effect of ±30% variation in the mass flow rate of water on $T_{c,out}$, U and ε is shown in the following figures. The methanol exit temperature and effectiveness of heat exchanger is observed to vary linearly with change in water mass flow rate. The overall heat transfer coefficient also increases with increase in the water mass flow rate as a consequence of increase in the convection heat transfer coefficient.

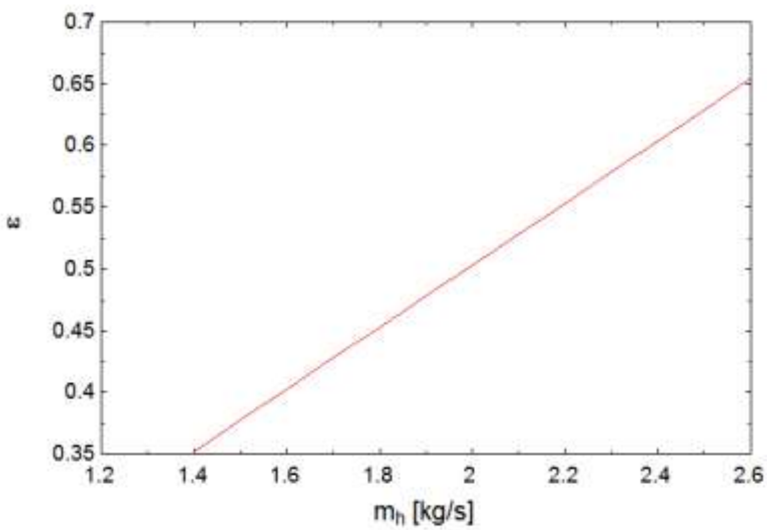
The capacity ratio '*c*' is always lower than unity except for the lowest water mass flow rate of 1.4 kg/s. In this case the heat capacity of methanol is more than that of the water.



Variation of the methanol exit temperature witch change in water mass flow rate.



Variation of overall heat transfer coefficient with change in water mass flow rate.



Variation of the heat exchanger effectiveness with change in water mass flow rate

Discussion In such problems multivariable optimization techniques can be used to decide upon the most optimum water flow rate that will give higher heat transfer rates with maximum economy.

11-118 Ethylene glycol flows over a staggered tube bank to heat the engine oil flowing through tubes. For the known geometry of the tube bank determine the mass flow rate of ethylene glycol and the number of tube rows.

Assumption **1** Steady state conditions exist. **2** Heat exchanger is well insulated. **3** Properties are constant. **4** No fouling inside the heat exchanger.

Properties Evaluate the properties of oil at an average temperature of (10 + 70)°C /2 = 40°C from Table A-13: c_p = 1964 J/kg·K

Analysis (a) From energy balance between ethylene glycol and engine oil we get,

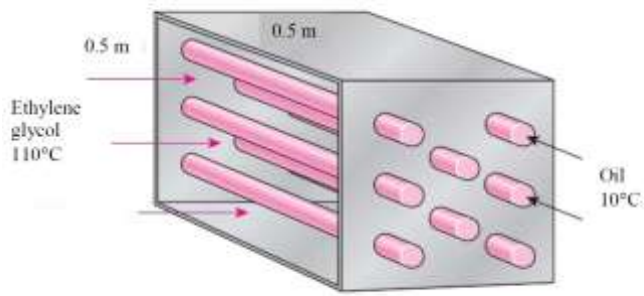
Heat lost by ethylene glycol = Heat gained by engine oil.

$$\dot{Q} = \dot{m}_h c_{ph}(T_{c,in} - T_{c,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\therefore \dot{m}_h = \frac{\dot{m}_c c_{pc}(T_{c,out} - T_{c,in})}{c_{ph}(T_{h,in} - T_{h,out})} \rightarrow \dot{m}_h = \frac{(4.05\ \text{kg/s})(1964\ \text{J/kg}\cdot\text{K})(70 - 10)^\circ\text{C}}{(2742\ \text{J/kg}\cdot\text{K})(110 - 90)^\circ\text{C}} = \mathbf{8.7\ kg/s}$$

Thus the mass flow rate of ethylene glycol is **8.7 kg/s**.

(b) For the given tube bank geometry,



$$S_D = \sqrt{S_L^2 + (S_T/2)^2} = \sqrt{0.035^2 + 0.0175^2} = 0.0391$$

For staggered arrangement since $S_D > S_T$, we have $2A_D > A_T$.

Thus the maximum velocity across staggered tube banks is calculated from Equation (7-40).

$$V_{\max} = \frac{S_T}{S_T - D_o} V \rightarrow V_{\max} = \frac{S_T}{S_T - D_o} \frac{\dot{m}_h}{\rho A}$$

$$V_{\max} = \frac{0.035}{(0.035 - 0.025)} \frac{8.7\ \text{kg/s}}{(1062\ \text{kg/m}^3)(0.5 \times 0.5)\text{m}^2} = 0.114\ \text{m/s}$$

Reynolds number based on maximum velocity is,

$$\text{Re}_D = \frac{\rho V_{\max} D_o}{\mu} = \frac{(1062\ \text{kg/m}^3)(0.114\ \text{m/s})(0.025\ \text{m})}{2.499 \times 10^{-3}\ \text{kg/m}\cdot\text{s}} = 1211$$

For staggered arrangement we initially assume that the number of tube rows are greater than 16. Thus from Table 7-2 we get,

$$Nu_D = 0.35\left(\frac{S_T}{S_L}\right)^{0.2} \text{Re}_D^{0.6}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25} \quad \textit{where } S_T = S_L$$

$$\therefore Nu_D = 0.35(1211)^{0.6}(26.12)^{0.36}(26.12/96.97)^{0.25} = 57.76$$

Hence the heat transfer coefficient on the ethylene glycol side is,

$$h_o = \frac{Nu_D k}{D_o} = \frac{57.76(0.262\ \text{W/m}\cdot\text{K})}{0.025\text{m}} = 605.3\ \text{W/m}^2\cdot\text{K}$$

The overall heat transfer coefficient is,

$$\frac{1}{U} = \frac{D_o}{h_i D_i} + \frac{D_o \ln(D_o/D_i)}{2k} + \frac{1}{h_o}$$

$$= \frac{0.025\text{m}}{(2500\ \text{W/m}^2\cdot\text{K})(0.023\text{m})} + \frac{(0.025\,m)\ln(0.025/0.023)}{2(250\ \text{W/m}\cdot\text{K})} + \frac{1}{605.3\ \text{W/m}^2\cdot\text{K}}$$

$$\therefore \frac{1}{U} = (4.34 \times 10^{-4} + 4.169 \times 10^{-6} + 1.652 \times 10^{-3})\ \text{m}^2\cdot\text{K/W}$$

The overall heat transfer coefficient is

$$U = 478.4\ \text{W/m}^2 \cdot \text{K}$$

The heat capacity of ethylene glycol is

$$C_h = \dot{m}_h c_{ph} = (8.7\ \text{kg/s})(2742\ \text{J/kg}\cdot\text{K}) = 23855.4\ \text{W/K} \rightarrow \text{C}_{\text{max}}$$

The heat capacity of engine oil is

$$\text{C}_\text{c} = \dot{\text{m}}_\text{c}\text{c}_\text{pc} = (4.05\,\text{kg/s})(1964\ \text{J/kg}\cdot\text{K}) = 7954.2\,\text{W/K} \rightarrow \text{C}_\text{min}$$

Thus the capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{7954.2}{23855.4} = 0.333$$

The effectiveness of heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{(70-10)^o C}{(110-10)^o C} = 0.6$$

The number of transfer units (NTU) of heat exchanger is calculated from Table 11-5 with C_{max} mixed and C_{min} unmixed as

$$NTU = -\ln\left[1 + \frac{\ln(1-\varepsilon\, c)}{c}\right] = -\ln\left[1 + \frac{\ln(1-0.6\times 0.333)}{0.333}\right] = 1.106$$

From the definition of NTU, we find the surface area of the heat exchanger as,

$$A_s = \frac{NTU\, C_{\min}}{U} = \frac{(1.106)\ (7954.2\,\text{W/K})}{478.4\,\text{W/m}^2\cdot\text{K}} = 18.38\,\text{m}^2$$

Also the surface area of heat exchanger is expressed as

$$A_s = \pi D_o L \times N_L \times N_T$$

The number of transverse rows are

$$N_T = 0.5 / S_T = 14$$

Thus the number of tube rows are

$$N_L = \frac{A_s}{\pi D_o L N_T} = \frac{18.38\,\text{m}^2}{\pi(0.025\,\text{m})(0.5\,\text{m})(14)} \approx \mathbf{33}$$

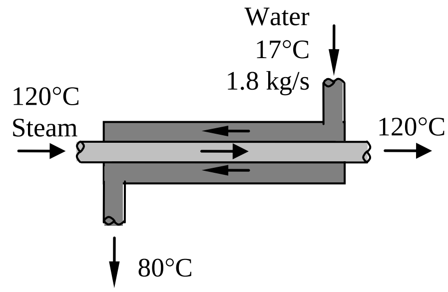
Discussion Initial assumption of number tube rows more than 16 is correct. In this problem the convection heat transfer coefficient was assumed. However, it is recommended to calculate the 'h' value for oil and verify if the assumption is correct.

11-119 Water is heated by steam condensing in a condenser. The required length of the tube is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of vaporization of water at 120°C is given to be 2203 kJ/kg.

Analysis (*a*) The temperature differences between the steam and the water at the two ends of the condenser are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 120°\text{C} - 80°\text{C} = 40°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 120°\text{C} - 17°\text{C} = 103°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{40-103}{\ln(40/103)} = 66.61°\text{C}$$

The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (1.8\,\text{kg/s})(4.18\,\text{kJ/kg.°C})(80°\text{C} - 17°\text{C}) = 474.0\,\text{kW}$$

The surface area of heat transfer is

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{474.0\,\text{kW}}{(0.7\,\text{kW/m}^2.°\text{C})(66.61°\text{C})} = 10.17\,\text{m}^2$$

The length of tube required then becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{10.17\,\text{m}^2}{\pi(0.025\,\text{m})} = \mathbf{129\,m}$$

(*b*) The maximum rate of heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (1.8\,\text{kg/s})(4.18\,\text{kJ/kg.°C})(120°\text{C} - 17°\text{C}) = 775.0\,\text{kW}$$

Then the effectiveness of this heat exchanger becomes

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{474.0\,\text{kW}}{775.0\,\text{kW}} = 0.6116$$

The NTU of this heat exchanger is determined using the relation in Table 11-5 to be

$$\text{NTU} = -\ln(1-\varepsilon) = -\ln(1-0.6116) = 0.9457$$

The surface area is

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{NTU\,C_{\min}}{U} = \frac{(0.9457)(1.8\,\text{kg/s})(4.18\,\text{kJ/kg.°C})}{0.7\,\text{kW/m}^2.°\text{C}} = 10.16\,\text{m}^2$$

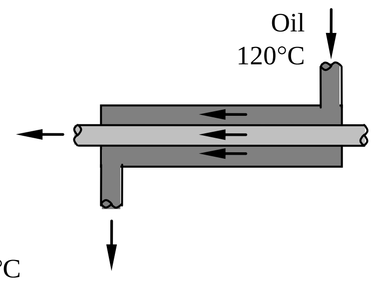
Finally, the length of tube required is

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{10.16\,\text{m}^2}{\pi(0.025\,\text{m})} = \mathbf{129\,m}$$

11-120 Ethanol is vaporized by hot oil in a double-pipe parallel-flow heat exchanger. The outlet temperature and the mass flow rate of oil are to be determined using the LMTD and NTU methods.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of oil is given to be 2.2 kJ/kg.°C. The heat of vaporization of ethanol at 78°C is given to be 846 kJ/kg.



Analysis (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}h_{fg} = (0.03\,\text{kg/s})(846\,\text{kJ/kg}) = 25.38\,\text{kW}$$

The log mean temperature difference is

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow \Delta T_{lm} = \frac{\dot{Q}}{UA_s} = \frac{25{,}380\text{ W}}{(320\text{ W/m}^2.°\text{C})(6.2\,\text{m}^2)} = 12.79°\text{C}$$

The outlet temperature of the hot fluid can be determined as follows

$$\Delta T_1 = T_{h,in} - T_{c,in} = 120°\text{C} - 78°\text{C} = 42°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,out} = T_{h,out} - 78°\text{C}$$

and $$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{42-(T_{h,out}-78)}{\ln[42/(T_{h,out}-78)]} = 12.79°\text{C}$$

whose solution is

$$T_{h,out} = \mathbf{79.8°C}$$

Then the mass flow rate of the hot oil becomes

$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{h,in} - T_{h,out})} = \frac{25{,}380\text{ W}}{(2200\text{ J/kg.°C})(120°\text{C} - 79.8°\text{C})} = \mathbf{0.287\,kg/s}$$

(*b*) The heat capacity rate $C = \dot{m}c_p$ of a fluid condensing or evaporating in a heat exchanger is infinity, and thus

$$c = C_{\min}/C_{\max} = 0\,.$$

The effectiveness in this case is determined from $\varepsilon = 1 - e^{-NTU}$

where $$NTU = \frac{UA_s}{C_{\min}} = \frac{(320\text{ W/m}^2.°\text{C})(6.2\,\text{m}^2)}{(\dot{m},\text{ kg/s})(2200\text{ J/kg.°C})}$$

and $$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in})$$

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_{\min}(T_{h,in}-T_{c,in})}{C_{\min}(T_{h,in}-T_{c,in})} = \frac{120 - T_{h,out}}{120-78}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) = 25{,}380\text{ W}$$
$$\dot{Q} = \dot{m}\times 2200(120 - T_{h,out}) = 25{,}380\text{ W} \qquad (1)$$

Also $$\frac{120 - T_{h,out}}{120-78} = 1 - e^{-\frac{6.2\times 320}{\dot{m}\times 2200}} \qquad (2)$$

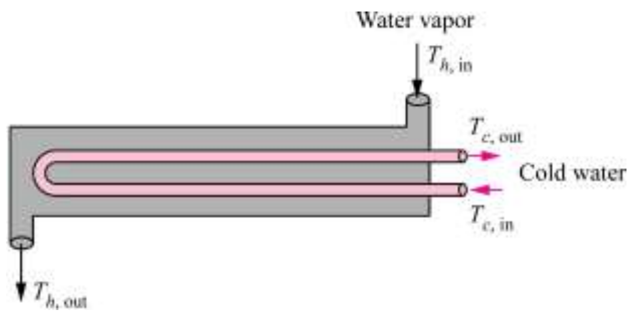
Solving (1) and (2) simultaneously gives

$$\dot{m}_h = \mathbf{0.287\,kg/s} \quad \text{and} \quad T_{h,out} = \mathbf{79.8°C}$$

11-121 Saturated water vapor condenses in a 1-shell and 2-tube heat exchanger, (*a*) the heat transfer effectiveness, (*b*) the outlet temperature of the cold water, and (*c*) the heat transfer rate for the heat exchanger are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the cold water is given to be c_{pc} = 4179 J/kg·K.



Analysis (*a*) The minimum heat capacity rate is from the cold fluid, since for the hot fluid,

$$C_h = C_{\max} \to \infty$$

So, we have

$$C_c = C_{\min} = \dot{m}_c c_{pc} = (0.5\text{ kg/s})(4179\text{ J/kg}\cdot\text{K})$$
$$= 2090\text{ W/K}$$

The heat capacity ratio in condensation process is

$$c = \frac{C_c}{C_h} = \frac{C_{\min}}{C_{\max}} \to 0$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(2000\text{ W/m}^2\cdot\text{K})(0.5\text{ m}^2)}{2090\text{ W/K}} = 0.4785$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\varepsilon = 1 - \exp(-\text{NTU}) = 1 - \exp(-0.4785) = \mathbf{0.380}$$

(*b*) The outlet temperature of the cold water can be determined using

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_c(T_{h,\text{in}} - T_{c,\text{in}})}$$

$$T_{c,\text{out}} = \varepsilon(T_{h,\text{in}} - T_{c,\text{in}}) + T_{c,\text{in}} = (0.380)(100 - 15)°\text{C} + 15°\text{C} = \mathbf{47.3°C}$$

(*c*) The heat transfer rate for the heat exchanger is

$$\dot{Q} = C_c(T_{c,\text{out}} - T_{c,\text{in}}) = (2090\text{ W/K})(47.3 - 15)\text{ K} = \mathbf{6.75 \times 10^4\ W}$$

Discussion The rate of heat transfer in the heat exchanger can also be calculated using

$$\dot{Q} = C_{\min}\varepsilon(T_{h,\text{in}} - T_{c,\text{in}})$$

11-122 Steam is condensed by cooling water in a shell-and-tube heat exchanger. The rate of heat transfer and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform. **5** The thickness of the tube is negligible.

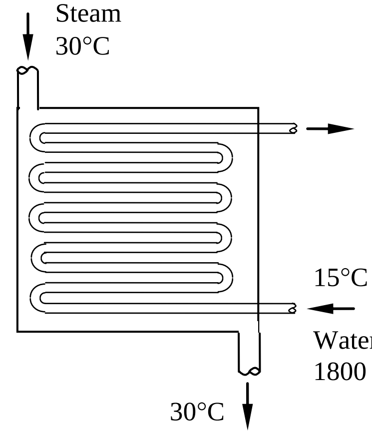
Properties The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of condensation of steam at 30°C is given to be 2430 kJ/kg.

Analysis (*a*) The heat capacity rate of a fluid condensing in a heat exchanger is infinity. Therefore,

$$C_{\min} = C_c = \dot{m}_c c_{pc} = (1800/3600\ \text{kg/s})(4.18\ \text{kJ/kg.°C}) = 2.090\ \text{kW/°C}$$

and

$$c = 0$$



Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (2.090\ \text{kW/°C})(30°\text{C} - 15°\text{C}) = 31.35\ \text{kW}$$

and

$$A_s = 8n\pi DL = 8 \times 50\pi(0.015\ \text{m})(2\ \text{m}) = 37.70\ \text{m}^2$$

The NTU of this heat exchanger

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(3\ \text{kW/m}^2.°\text{C})(37.70\ \text{m}^2)}{2.090\ \text{kW/°C}} = 54.11$$

Then the effectiveness of this heat exchanger corresponding to c = 0 and NTU = 54.11 is determined using the proper relation in Table 11-5

$$\varepsilon = 1 - \exp(-\text{NTU}) = 1 - \exp(-54.11) = 1$$

Then the actual heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (1)(31.35\ \text{kW}) = \mathbf{31.35 kW}$$

(*b*) Finally, the rate of condensation of the steam is determined from

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{31.35\ \text{kJ/s}}{2431\ \text{kJ/kg}} = \mathbf{0.0129 kg/s}$$

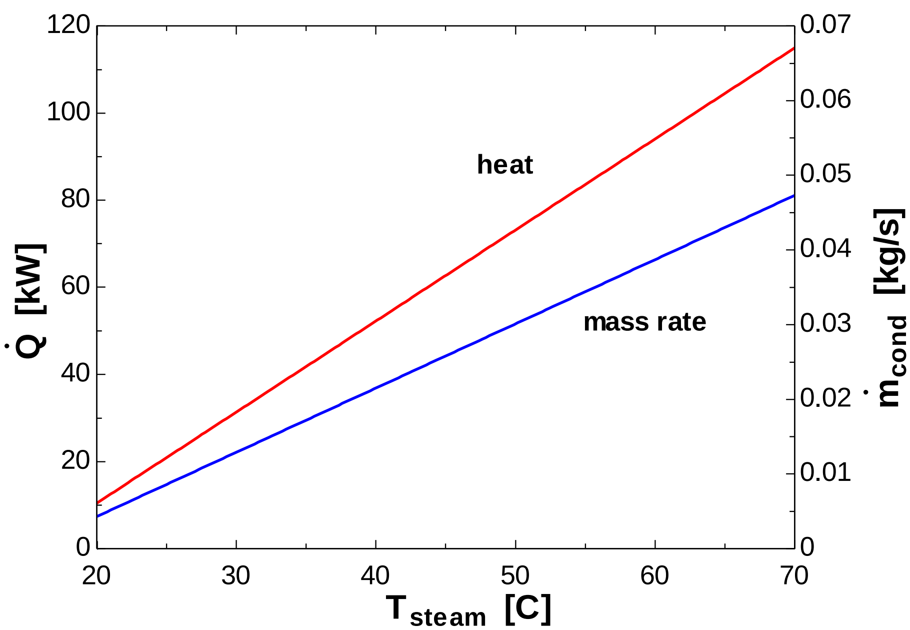
11-123 Prob. 11-122 is reconsidered. The effects of the condensing steam temperature and the tube diameter on the rate of heat transfer and the rate of condensation of steam are to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

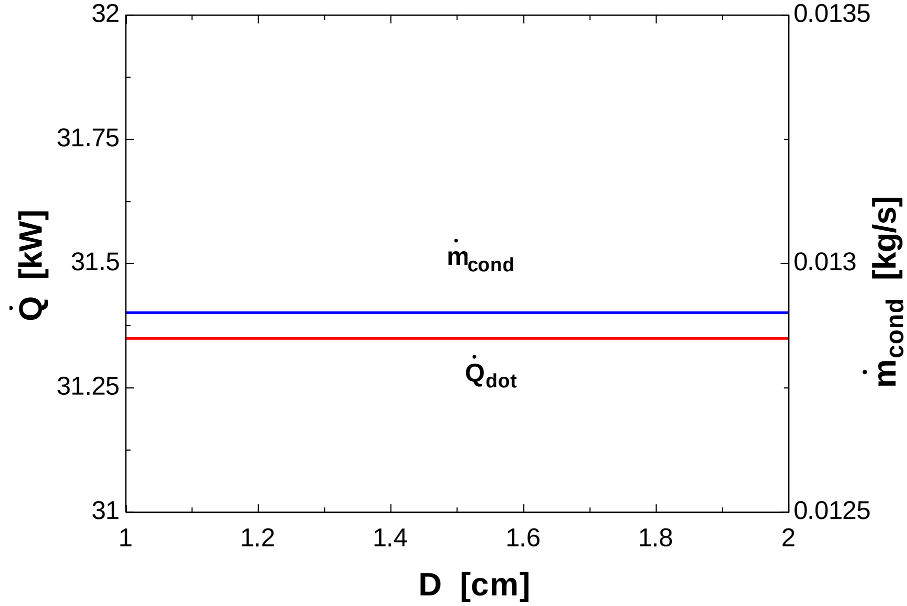
```
"GIVEN"
N_pass=8
N_tube=50
T_steam=30 [C]
h_fg_steam=2431 [kJ/kg]
T_w_in=15 [C]
m_dot_w=1800[kg/h]*Convert(kg/h, kg/s)
c_p_w=4.18 [kJ/kg-C]
D=1.5 [cm]
L=2 [m]
U=3 [kW/m^2-C]

"ANALYSIS"
"With EES, it is easier to solve this problem using LMTD method than NTU method. Below, we use NTU method. Both methods give the same results."
C_min=m_dot_w*c_p_w
c=0 "since the heat capacity rate of a fluid condensing is infinity"
Q_dot_max=C_min*(T_steam-T_w_in)
A=N_pass*N_tube*pi*D*L*Convert(cm, m)
NTU=(U*A)/C_min
epsilon=1-exp(-NTU) "from Table 11-4 of the text with c=0"
Q_dot=epsilon*Q_dot_max
Q_dot=m_dot_cond*h_fg_steam
```

T_{steam} [C]	$\dot{Q}$ [kW]	$\dot{m}_{cond}$ [kg/s]
20	10.45	0.0043
22.5	15.68	0.006451
25	20.9	0.008601
27.5	26.13	0.01075
30	31.35	0.0129
32.5	36.58	0.01505
35	41.8	0.0172
37.5	47.03	0.01935
40	52.25	0.0215
42.5	57.48	0.02365
45	62.7	0.0258
47.5	67.93	0.02795
50	73.15	0.0301
52.5	78.38	0.03225
55	83.6	0.0344
57.5	88.83	0.03655
60	94.05	0.0387
62.5	99.28	0.04085
65	104.5	0.043
67.5	109.7	0.04515
70	115	0.0473



D [cm]	$\dot{Q}$ [kW]	$\dot{m}_{cond}$ [kg/s]
1	31.35	0.0129
1.05	31.35	0.0129
1.1	31.35	0.0129
1.15	31.35	0.0129
1.2	31.35	0.0129
1.25	31.35	0.0129
1.3	31.35	0.0129
1.35	31.35	0.0129
1.4	31.35	0.0129
1.45	31.35	0.0129
1.5	31.35	0.0129
1.55	31.35	0.0129
1.6	31.35	0.0129
1.65	31.35	0.0129
1.7	31.35	0.0129
1.75	31.35	0.0129
1.8	31.35	0.0129
1.85	31.35	0.0129
1.9	31.35	0.0129
1.95	31.35	0.0129
2	31.35	0.0129



11-124E The hot water exiting the condenser is to be cooled by passing it through a heat exchanger immersed in large lake. Using $\varepsilon - NTU$ method, exit temperature of the water from immersed heat exchanger is to be determined.

Assumptions **1** Steady state conditions exist. **2** Fluid properties remain constant. **3** Lake water is an infinite medium.

Analysis Since the problem statement requires use of $\varepsilon - NTU$ method we first calculate the heat capacity rates of cold and hot fluids. The cold side i.e. lake water is an infinite medium and its temperature remain virtually unchanged. Thus the cold side heat capacity C_c is $C_{max,}$ since $C_{max} \to \infty$, then c = 0. The hot side heat capacity (condenser exit water) is

$$C_h = \dot{m}_h c_{ph} = (\rho A_c V)\, c_{ph}$$

$$\therefore C_h = (62.4\,\text{lbm/ft}^3)[\frac{\pi}{4}\left(\frac{1}{12}\right)^2 \text{ft}^2](9\,\text{ft/s}) \times 1(\text{Btu/lbm}\cdot{}^\circ\text{F}) = 3.06\,\text{Btu/s}\cdot{}^\circ\text{F}$$

From the definition of NTU we have

$$NTU = \frac{UA_s}{C_{\min}} = \frac{U\pi DL}{C_{\min}} = \frac{(250\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})[\pi\,(1/12)\text{ft}\;(500\,\text{ft})]}{3.06\,\text{Btu/s}\cdot{}^\circ\text{F}}\,\frac{1\text{h}}{3600\text{s}} = 2.97$$

Now form the equation in Table 11-4, the effectiveness is calculated as

$$\varepsilon = 1 - \exp(-NTU) = 1 - \exp(-2.97) = 0.948$$

The heat transfer rate is,

$$\dot{Q} = \varepsilon\;\dot{Q}_{\max} = \varepsilon\, C_{\min}\,(T_{h,in} - T_{c,in})$$

$$\therefore \dot{Q} = (0.948)\,(3.06\,\text{Btu/s}\cdot{}^\circ\text{F})(100 - 45)^\circ F = 159.55\,\text{Btu/s}$$

Thus from energy balance we can find the exit temperature of the water from immersed heat exchanger as,

$$\dot{Q} = C_h\,(T_{h,in} - T_{h,out}) = (3.06\,\text{Btu/s}\cdot{}^\circ\text{F}) \times (100 - T_{h,out})^\circ\text{F}$$

$$\therefore T_{h,out} = \mathbf{47.9\ {}^o F}$$

Discussion In case of the heat exchangers immersed in ponds or lakes, the immersion depth plays a key role in maintaining the fluid exit temperature. For shallow depths, the lake water temperature is subject to change with change in environmental conditions and hence influences the fluid exit temperature.

11-125 Saturated steam flows over tubes in a shell and tube heat exchanger at specified conditions. Effectiveness of the heat exchanger, length of the tube and the rate of steam condensation is to be determined.

Assumptions **1** Steady state conditions exist. **2** Fluid properties are constant. **3** Heat exchanger is well insulated.

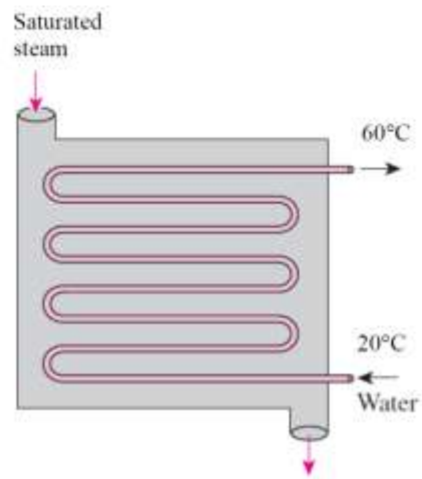
Properties The properties of water are evaluated at an average temperature of (60 + 20)°C/2 = 40°C from Table A-9: $\rho = 992.1\,\text{kg/m}^3$, $\mu = 0.653\times10^{-3}$ kg/m·s, $c_p = 4179$ J/kg·K , k = 0.631 W/m·K , and Pr = 4.32.

The properties of steam are calculated at saturation temperature corresponding to the given saturation pressure. At a saturation pressure of 270.1 kPa from Table A-9, the saturation temperature of the steam is 130°C. At 130°C, the enthalpy of vaporization is 2174 kJ/kg.

Analysis (a) Since the steam undergoes a condensation process the temperature of steam remains unchanged at 130 °C. Hence the C_{max} is infinity and thus C_{min} is for the cold fluid. Thus the capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} \to 0\,.$$

The effectiveness is calculated as



$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c\,(T_{c,out} - T_{c,in})}{C_{\min}\,(T_{h,in} - T_{c,in})} = \frac{60-20}{130-20} = \mathbf{0.363}$$

(b) C_{min} is calculated as

$$C_{\min} = \dot{m}_c c_{pc} = (0.25\,\text{kg/s})(4179\ \text{J/kg}\cdot\text{K}) = 1044.75\,\text{W/K}$$

For a phase change process, the number of transfer units (NTU) is calculated as

$$NTU = -\ln(1-\varepsilon) = -\ln(1-0.363) = 0.45$$

The number of transfer units is defined as

$$NTU = \frac{UA_s}{C_{\min}} = \frac{Un\pi DL}{C_{\min}}$$

Here we first need to calculate the overall heat transfer coefficient. For the given ... t

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu} = \frac{4(0.25\text{ kg/s})}{\pi\,(0.0125\text{ m})(0.653\times10^{-3}\text{ kg/m}\cdot\text{s})} = 38997$$

Since Re > 10,000, the flow is fully turbulent. Assuming the flow to be fully developed the Nusselt number can be determined from Dittus-Bolter equation:

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\ \text{Pr}^{0.4} = 0.023(38997)^{0.8}(4.32)^{0.4} = 194.42$$

The convection heat transfer coefficient is

$$h = \frac{Nu\,k}{D} = \frac{(194.42\,\text{W/m}^2\cdot\text{K})(0.631\text{W/m}\cdot\text{K})}{0.0125\text{ m}} = 9814.4\,\text{W/m}^2\cdot\text{K}$$

Thus the overall heat transfer coefficient is

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} + R_{f,i} + R_{f,o} = \frac{1}{9814.4\,\text{W/m}^2\cdot\text{K}} + \frac{1}{1500\,\text{W/m}^2\cdot\text{K}} + (0.00015\,\text{m}^2\cdot\text{K/W}) + (0.0001\text{ m}^2\cdot\text{K/W})$$

$$U = 981.8\ \ \text{W/m}^2\cdot\text{K}$$

Therefore the length of heat exchanger tube is calculated as

$$L = \frac{NTU\,C_{\min}}{U\,\pi\,D\,n} = \frac{0.45\ (1044.75\,\text{W/K})}{(981.8\,\text{W/m}^2\cdot\text{K})\,\pi\,(0.0125\,m)\,(4)} = \mathbf{3.04\ \ m}$$

(c) From the energy balance we have,

$$\dot{m}_h h_{fg} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\dot{m}_h = \frac{(0.25\,\text{kg/s})(4179\,\text{J/kg}\cdot\text{K})(60-20)^{\text{o}}\text{C}}{2174\times10^3\ \text{J/kg}} = \mathbf{0.0192\,kg/s = 69.2\,kg/h}$$

11-126 Cold water is used to condense saturated refrigerant R134a in a shell and tube heat exchanger. For a given overall heat transfer coefficient the heat exchanger area and mass flow rate of cooling water is to be determined. In case of fouling when the overall heat transfer coefficient drops down, new flow rate of cooling water for complete condensation of refrigerant is to be determined.

Assumptions **1** Steady state conditions exist. **2** Heat exchanger is well insulated. **3** Fluid properties remain constant. **4** Thermal resistance due to heat exchanger tube walls is negligible.

Properties Thermo physical properties of the cooling water are evaluated at an average temperature of (10 + 40)°C/2 = 25°C from Table A-9. $c_p = 4180\ \text{J/kg}\cdot\text{K}$.

Properties of refrigerant R134a are calculated at 1318.6 kPa from Table A-10.

$$T_{sat} = T_{h,in} = T_{h,out} = 50^o\text{C},\ h_{fg} = 151.8\times10^3\ \text{J/kg}\cdot\text{K}$$

Analysis (a) From energy balance we have,

The required rate of heat transfer for complete condensation is,

$$\dot{Q} = \dot{m}_h h_{fg} = (2.5\ \text{kg/s})(151.8\times10^3\ \text{J/kg}\cdot\text{K}) = 3.795\times10^5\ \text{W}$$

From heat balance, the heat lost by refrigerant = heat gained by cooling water.

$$\dot{Q} = \dot{m}_h h_{fg} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\therefore \dot{m}_c = \frac{\dot{m}_h h_{fg}}{c_{pc}(T_{c,out} - T_{c,in})} = \frac{3.795\times10^5\ \text{W}}{(4180\ \text{J/kg}\cdot\text{K})(40-10)^o\text{C}} = \mathbf{3.02\ kg/s}$$

The heat capacity of the cold water is

$$C_c = \dot{m}_c c_{pc} = (3.02\ \text{kg/s})(4180\ \text{J/kg}\cdot\text{K}) = 12624\ \text{W/K}$$

Since the saturated refrigerant undergoes a phase change process, $C_h \to \infty$ and hence the capacity ratio c = 0.

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{\dot{m}_h h_{fg}}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{3.795\times10^5\ \text{W}}{(12624\ \text{W/K})(50-10)^o C} = 0.75$$

Thus the number of transfer units (NTU) are

$$NTU = -\ln(1-\varepsilon) = -\ln(1-0.75) = 1.38$$

From the definition of NTU we find heat exchanger surface area as

$$A_s = \frac{NTU\, C_{\min}}{U} = \frac{1.38\,(12624\ \text{W/K})}{3500\ \text{W/m}^2\cdot\text{K}} = \mathbf{4.98\ m^2}$$

(b) In case of fouling, the new heat transfer coefficient is 2800 W/m^2·K (0.2 × 3500 W/m^2·K).

Based on this new heat transfer coefficient we have the rate of heat transfer as

$$\dot{Q} = UA_s\Delta T_{lm}$$

Now the logarithmic temperature difference is calculated as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 50 - 40 = 10^o C$$

and

$$\Delta T_2 = T_{h,out} - T_{c,in} = 50 - 10 = 40^o C$$

$$\Delta T_{lm} = \frac{10-40}{\ln(10/40)} = 21.64\ ^{\circ}\text{C}$$

Therefore the rate of heat transfer is

$$\dot{Q} = UA_s\Delta T_{lm} = (2800\ \text{W/m}^2\cdot\text{K})(4.98\text{m}^2)(21.64^{\circ}\text{C}) = 3.0174\times10^5\ \text{W}$$

Thus the reduction in heat transfer due to fouling is $(3.795\times10^5\ \text{W} - 3.0174\times10^5\ \text{W}) = 0.777\times10^5$ W and hence the refrigerant will not condense completely to saturated liquid at the heat exchanger exit. In order to overcome this situation, the mass flow rate of cooling water has to be increased to account for this heat transfer reduction. Thus the new mass flow rate of cooling water is,

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\therefore \dot{m}_c = \frac{\dot{Q}}{c_{pc}(T_{c,out} - T_{c,in})} = \frac{(3.795+0.777)\times10^5\ \text{W}}{(4182\ \text{J/kg}\cdot\text{K})(40-10)^{\circ}\text{C}} = \mathbf{3.64\,kg/s}$$

The mass flow rate of cooling water that accounts for fouling in the heat exchanger is **3.64 kg/s**.

Discussion Fouling in heat exchanger causes the mass flow rate of cooling water to be increased by about 21%. In case of large scale systems, this significant increase in mass flow rates will require additional pumping making the heat exchange process uneconomical.

Selection of the Heat Exchangers

11-127C In the case of automotive and aerospace industry, where weight and size considerations are important, and in situations where the space availability is limited, we choose the smaller heat exchanger.

11-128C The first thing we need to do is determine the life expectancy of the system. Then we need to evaluate how much the larger will save in pumping cost, and compare it to the initial cost difference of the two units. If the larger system saves more than the cost difference in its lifetime, it should be preferred.

11-129C 1) Calculate heat transfer rate, 2) select a suitable type of heat exchanger, 3) select a suitable type of cooling fluid, and its temperature range, 4) calculate or select U, and 5) calculate the size (surface area) of heat exchanger

11-130 Oil is to be cooled by water in a heat exchanger. The heat transfer rating of the heat exchanger is to be determined and a suitable type is to be proposed.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the oil is given to be 2.2 kJ/kg.°C.

Analysis The heat transfer rate of this heat exchanger is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (13\,\text{kg/s})(2.2\,\text{kJ/kg.°C})(120°\text{C} - 50°\text{C}) = \mathbf{2002\,kW}$$

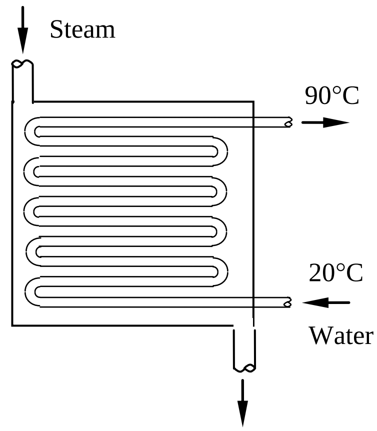
We propose a compact heat exchanger (like the car radiator) if air cooling is to be used, or a tube-and-shell or plate heat exchanger if water cooling is to be used.

11-131 Water is to be heated by steam in a shell-and-tube process heater. The number of tube passes need to be used is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the water is given to be 4.19 kJ/kg.°C.

Analysis The mass flow rate of the water is



$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\dot{m} = \frac{\dot{Q}}{c_{pc}(T_{c,out} - T_{c,in})} = \frac{600\text{ kW}}{(4.19\text{ kJ/kg.°C})(90°\text{C} - 20°\text{C})} = 2.046\text{ kg/s}$$

The total cross-section area of the tubes corresponding to this mass flow rate is

$$\dot{m} = \rho V A_c \rightarrow A_c = \frac{\dot{m}}{\rho V} = \frac{2.046\text{ kg/s}}{(1000\text{ kg/m}^3)(3\text{ m/s})} = 6.82\times10^{-4}\text{ m}^2$$

Then the number of tubes that need to be used becomes

$$A_s = n\frac{\pi D^2}{4} \longrightarrow n = \frac{4A_s}{\pi D^2} = \frac{4(6.82\times10^{-4}\text{ m}^2)}{\pi(0.01\text{ m})^2} = 8.68 \cong \mathbf{9}$$

Therefore, we need to use at least 9 tubes entering the heat exchanger.

11-132 Prob. 11-131 is reconsidered. The number of tube passes as a function of water velocity is to be plotted.

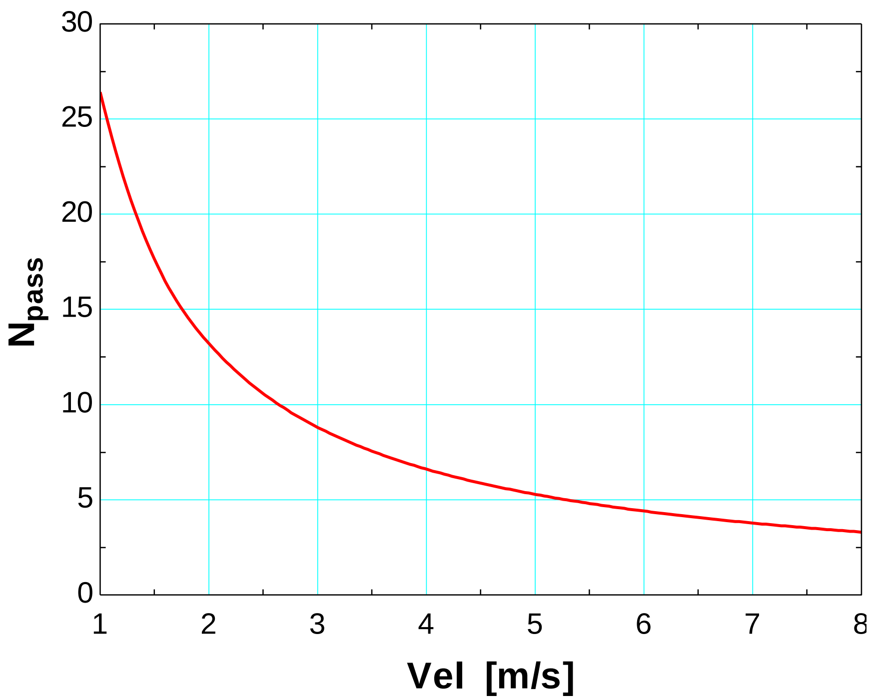
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
c_p_w=4.19 [kJ/kg-C]
T_w_in=20 [C]
T_w_out=90 [C]
Q_dot=600 [kW]
D=0.01 [m]
Vel=3 [m/s]

"PROPERTIES"
rho=density(water, T=T_ave, P=100)
T_ave=1/2*(T_w_in+T_w_out)

"ANALYSIS"
Q_dot=m_dot_w*c_p_w*(T_w_out-T_w_in)
m_dot_w=rho*A_c*Vel
A_c=N_pass*pi*D^2/4
```

Vel [m/s]	N_{pass}
1	26.42
1.5	17.62
2	13.21
2.5	10.57
3	8.808
3.5	7.55
4	6.606
4.5	5.872
5	5.285
5.5	4.804
6	4.404
6.5	4.065
7	3.775
7.5	3.523
8	3.303

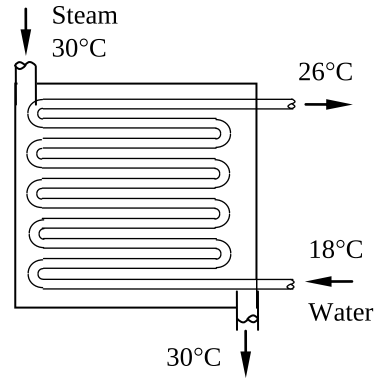


11-133 Cooling water is used to condense the steam in a power plant. The total length of the tubes required in the condenser is to be determined and a suitable HX type is to be proposed.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of the water is given to be 4.18 kJ/kg.°C. The heat of condensation of steam at 30°C is given to be 2431 kJ/kg.

Analysis The temperature differences between the steam and the water at the two ends of condenser are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 30°\text{C} - 26°\text{C} = 4°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 30°\text{C} - 18°\text{C} = 12°\text{C}$$

and the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{4-12}{\ln(4/12)} = 7.282°\text{C}$$

The heat transfer surface area is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{500\times 10^6 \text{ W}}{(3500 \text{ W/m}^2.°\text{C})(7.282°\text{C})} = 19{,}618\,\text{m}^2$$

The total length of the tubes required in this condenser then becomes

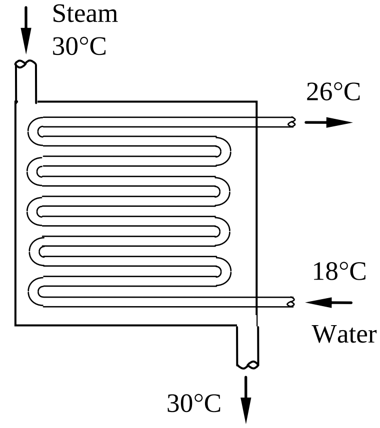
$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{19{,}618\,\text{m}^2}{\pi(0.02\,\text{m})} = 312{,}230\text{ m} = \mathbf{312\,km}$$

A multi-pass shell-and-tube heat exchanger is suitable in this case.

11-134 Cold water is heated by hot water in a heat exchanger. The net rate of heat transfer and the heat transfer surface area of the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.



Analysis The temperature differences between the steam and the water at the two ends of condenser are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 30°\text{C} - 26°\text{C} = 4°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 30°\text{C} - 18°\text{C} = 12°\text{C}$$

and the logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{4-12}{\ln(4/12)} = 7.28°\text{C}$$

The heat transfer surface area is

$$\dot{Q} = UA_s \Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{50\times 10^6 \text{ W}}{(3500 \text{ W/m}^2.°\text{C})(7.28°\text{C})} = 1962 \text{ m}^2$$

The total length of the tubes required in this condenser then becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{1962 \text{ m}^2}{\pi(0.02 \text{ m})} = 31{,}231 \text{ m} = \mathbf{31.2 km}$$

A multi-pass shell-and-tube heat exchanger is suitable in this case.

11-135 Petroleum based organic vapor is to be cooled inside a shell and tube heat exchanger. For the specified flow rates of each fluid, the number of tubes is to be found.

Assumptions **1** Steady state conditions exist. **2** Heat exchanger is insulated. **3** Negligible thermal resistance due to pipe wall thickness and negligible fouling. **4** Constant fluid properties.

Analysis From the energy balance between the organic vapor and cooling water we have

Heat lost by organic vapor = Heat gained by cooling water

$$\dot{Q} = \dot{m}_h h_{fg} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$\therefore (5\,\text{kg/s})(580\times10^3\ \text{J/kg}) = (25\,\text{kg/s})(4187\,\text{J/kg}\cdot\text{K})(T_{c,out} - 15)^\circ\text{C} = 29\times10^5\ \text{W}$$

Therefore the exit temperature of the cooling water is

$$T_{c,out} = 42.70\ ^\circ\text{C}$$

Now the logarithmic temperature difference is calculated as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 75 - 42.7 = 32.3^o C$$

and

$$\Delta T_2 = T_{h,out} - T_{c,in} = 75 - 15 = 60^o C$$

$$\Delta T_{lm} = \frac{32.3 - 60}{\ln(32.3/60)} = 44.73\ ^\circ\text{C}$$

The heat transfer rate is calculated as

$$\dot{Q} = \dot{m}_h h_{fg} = UA_s\Delta T_{lm} = U(n\pi DL)\Delta T_{lm}$$

$$29\times10^5\ \text{W} = (550\,\text{W/m}^2\cdot\text{K})\,\text{n}\,\pi\ (0.018\,\text{m})(5\,\text{m})(44.73^\circ\text{C})$$

$$\therefore n = \mathbf{417\ tubes}$$

11-136E Hot water flowing through the shell side at known mass flow rate is cooled by water flowing through tubes at specified mass flow rate and temperature. For the given space constraint, number of tube passes, number of tubes per pass and the length of each tube are to be determined.

Assumptions **1** Steady state operating conditions exist. **2** Heat exchanger is well insulated. **3** Fluid properties are constant. **4** No fouling inside heat exchanger.

Properties The specific heats of water on shell side and tube side are taken to be 1 Btu/lbm·°F from Table A-9E since for the range of temperatures in this problem the specific heat of water is almost constant and close to 1 Btu/lbm·°F. Also from Table A-9E the density of water on tube side is evaluated at an average temperature of (110 +80)°F/2 = 95°F is 62.06 lbm/ft^3.

Analysis As given in the problem statement we first calculate the tube length based on one tube pass and check if it satisfies the given tube length constraint. The exit temperature of the hot water is calculated by doing an energy balance. The heat gained by cold water is equal to the heat lost by hot water. Thus we have,

$$\dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$(30000\ lbm/h)(1\,\text{Btu/lbm}\cdot{}^\circ\text{F})(175 - T_{h,out})^\circ F = (40000\ \text{lbm/h})(1\,\text{Btu/lbm}\cdot{}^\circ\text{F})(110-80)^\circ\text{F}$$

$$\therefore T_{h,out} = 135^\circ\text{F}$$

The total heat transfer within the heat exchanger is,

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) = (40000\ \text{lbm/h})(1\,\text{Btu/lbm}\cdot{}^\circ\text{F})(110-80)^\circ\text{F} = 12\times10^5\ \text{Btu/h}$$

The heat transfer within a heat exchanger can also be calculated using overall heat transfer coefficient and the log mean temperature difference.

$$\dot{Q} = UA_s\Delta T_{lm}$$

where

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 175 - 110 = 65^o F$$

and $\Delta T_2 = T_{h,out} - T_{c,in} = 135 - 80 = 55^o F$

$$\Delta T_{lm} = \frac{65-55}{\ln(65/55)} = 59.9\ {}^\text{o}\text{F}$$

Thus the surface area of the heat exchanger is found to be

$$A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{12\times10^5\ \text{Btu/h}}{(220\,\text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})(59.9^\circ\text{F})} = 91.06\,\text{ft}^2$$

This is the total surface area of the heat exchanger based on 'n' number of tubes. We first need to find number of tubes in the heat exchanger based on the inside pipe diameter of the tubes and average water velocity.

$$\dot{m}_c = \rho A_c V \Rightarrow A_c = \frac{40000\,\text{lbm/h})}{(62.06\,\text{lbm/ft}^3)(1.5\text{ft/s})}\times\frac{1\text{h}}{3600\text{s}} = 0.1193\,\text{ft}^2$$

This cross sectional area is the total cross sectional area for 'n' tubes. Thus the number of tubes are,

$$A_c = n\frac{\pi D_i^2}{4} \Rightarrow n = \frac{4\,(0.1193\,\text{ft}^2)}{\pi\,(0.75/12)^2\ ft^2} = 38.88$$

Thus for $n \approx 39$ tubes and surface area of 91.06 ft^2, the length of each tube is calculated as,

$$A_s = n\pi D_i L \Rightarrow L = \frac{A_s}{n\pi D_i} = \frac{91.06\,\text{ft}^2}{39\pi\,(0.75/12)\text{ft}} = \mathbf{11.9\,ft}$$

The length of tube is greater than the given length constraint of 8 ft. Hence it is required to use more than one tube pass. For multiple pass heat exchangers we need to find the correction factor 'F' that accounts for the reduction in LMTD during each pass. The correction factor 'F' requires calculation of temperature ratios P and R from Equations 11-27 and 11-28, respectively.

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{110 - 80}{175 - 80} = 0.316 \quad \text{and} \quad R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{175 - 135}{110 - 80} = 1.33$$

Thus from Figure 11-19(a) we get

$F = 0.95$.

Based on this correction factor, the total surface area of the tubes is recalculated as,

$$A_s = \frac{\dot{Q}}{UF\Delta T_{lm}} = \frac{12 \times 10^5 \text{ Btu/h}}{(220 \text{ Btu/h} \cdot \text{ft}^2 \cdot °\text{F})(0.95)(59.9°\text{F})} = 95.85 \text{ ft}^2$$

For two tube pass heat exchanger, the length of each tube per pass is calculated as,

$$A_s = 2n\pi D_i L \Rightarrow L = \frac{A_s}{2n\pi D_i} = \frac{95.85 \text{ ft}^2}{2(39)\pi(0.75/12)\text{ft}} = 6.26 \text{ ft}$$

This tube length is now within the given space constraints. Hence the final heat exchanger design specifications are,

Number of tubes per pass = **39**

Number of tube pass = **2**

Length of tube per pass = **6.26 ft**

11-137 Saturated liquid benzene is to be cooled using cold water at 15°C. For the given value of overall heat transfer coefficient, the surface area of the heat exchanger is to be determined for four different configurations.

Assumptions **1** Steady state conditions exist. **2** Heat exchanger is well insulated. **3** Fluid properties remain constant. **4** There is no fouling inside the heat exchanger.

Analysis Heat balance between saturated liquid benzene solution and cooling water gives,

Heat lost by liquid benzene = Heat gained by water

$$\dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

$$T_{c,out} = \frac{\dot{m}_h c_{ph}(T_{h,in} - T_{h,out})}{\dot{m}_c c_{pc}} + T_{c,in} = \frac{(5\,\text{kg/s})\,(1839\ \text{J/kg}\cdot\text{K})\,(75 - 45)^{\circ}\text{C}}{3.5(\text{kg/s}) \times 4187(\text{J/kg}\cdot\text{K})} + 15^{\circ}\text{C} = 33.82^{\circ}\text{C}$$

In order to use effectiveness-NTU method we first need to determine the heat capacity rates, capacity ratio and effectiveness of the heat exchanger.

The heat capacity rate of cold fluid (water) is

$$C_c = \dot{m}_c c_{pc} = (3.5\,\text{kg/s})(4187\ \text{J/kg}\cdot\text{K}) = 14654.5\ \text{W/K}$$

The heat capacity rate of the hot fluid (liquid benzene) is

$$C_h = \dot{m}_h c_{ph} = (5\,\text{kg/s})(1839\,\text{J/kg}\cdot\text{K}) = 9195\ \text{W/K}$$

Thus the capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{9195}{14654.5} = 0.627$$

(a) For parallel flow arrangement,

$$\therefore\ \varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{75-45}{75-15} = 0.5$$

Therefore, for the known values of effectiveness and capacity ratio we can find the number of transfer units (NTU) from the relation given in Table 11-5.

$$NTU = -\frac{\ln\left[1-\varepsilon(1+c)\right]}{1+c} = -\frac{\ln\left[1-0.5(1+0.627)\right]}{1+0.627} = 1.032$$

From the definition of NTU we find the surface area as

$$A_s = \frac{NTU\,C_{\min}}{U} = \frac{1.032\,(9195\,\text{W/K})}{750\,\text{W/m}^2\cdot\text{K}} = \mathbf{12.65\,m^2}$$

Thus for a parallel flow arrangement with 12.65 m^2 surface area, the tube length is

$$L = \frac{A_s}{\pi D} = \frac{12.65(\text{m}^2)}{\pi \times 0.02(\text{m})} = 201.33\,\text{m}$$

(b) For counter flow arrangement, with ε = 0.5 from part (a), the number of transfer units (NTU) using the relation from Table 11-5 is found to be,

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon-1}{c\varepsilon-1}\right) = \frac{1}{0.627-1}\ln\left(\frac{0.5-1}{0.627\times 0.5-1}\right) = 0.85$$

From the definition of NTU we find the surface area as

$$A_s = \frac{NTU\,C_{\min}}{U} = \frac{0.85\,(9195\,\text{W/K})}{750\,\text{W/m}^2\cdot\text{K}} = \mathbf{10.42\,m^2}$$

Thus for a counter flow arrangement with 10.42 m^2 surface area, the tube length is

$$L = \frac{A_s}{\pi D} = \frac{10.42(\text{m}^2)}{\pi \times 0.02(\text{m})} = 165.85\ \text{m}$$

(c) For two shell passes and 40 tube passes, the calculated effectiveness of 0.5 from part (a) is for 2 shell passes.

$$\therefore \varepsilon_2 = 0.5$$

Now using expressions from Table 11-5 we have

$$F = \left(\frac{c\varepsilon_2 - 1}{\varepsilon_2 - 1}\right)^{1/2} = \left(\frac{0.627 \times 0.5 - 1}{0.5 - 1}\right)^{0.5} = 1.172$$

The effectiveness of heat exchanger in case of one shell pass would be

$$\varepsilon_1 = \frac{F - 1}{F - c} = \frac{1.172 - 1}{1.172 - 0.627} = 0.315$$

Based on this effectiveness for 1 shell pass we calculate the NTU_1 for 1 shell and 40 tube passes as follows.

$$NTU_1 = -\frac{1}{\sqrt{1+c^2}} \ln\left(\frac{2/\varepsilon_1 - 1 - c - \sqrt{1+c^2}}{2/\varepsilon_1 - 1 - c + \sqrt{1+c^2}}\right) = -\frac{1}{\sqrt{1+0.627^2}} \ln\left(\frac{2/0.315 - 1 - 0.627 - \sqrt{1+0.627^2}}{2/0.315 - 1 - 0.627 + \sqrt{1+0.627^2}}\right) = 0.433$$

Since for multiple passes, $NTU_2 = n(NTU_1)$ i.e., NTU is assumed to be distributed equally during each pass.

For two shell pass we have

$NTU_2 = 2\ (0.433) = 0.866.$

From the definition of NTU we find the surface area as

$$A_s = \frac{NTU_2\ C_{\min}}{U} = \frac{0.866\ (9195\ \text{W/K})}{750\ \text{W/m}^2 \cdot \text{K}} = \mathbf{10.62\ m^2}$$

Thus for a two shell and 40 tube pass flow arrangement with 10.62 m^2 surface area, the tube length is

$$L = \frac{A_s}{n \times \pi D} = \frac{10.62(\text{m}^2)}{\pi \times 0.02(\text{m}) \times 40} = 4.22\ \text{m}$$

(d) For a cross flow heat exchanger with liquid benzene (mixed) and cooling water (unmixed), from Table 11-5 and ε = 0.5 from part (a), we get,

$$NTU = -\frac{\ln[c\ln(1-\varepsilon)+1]}{c} = -\frac{\ln[0.627\ln(1-0.5)+1]}{0.627} = 0.909$$

From the definition of NTU we find the surface area as

$$A_s = \frac{NTU\ C_{\min}}{U} = \frac{0.909\ (9195\ \text{W/K})}{750\ \text{W/m}^2 \cdot \text{K}} = \mathbf{11.14\ m^2}$$

Discussion Although the heat exchanger surface area in case of counter flow arrangement is less than the shell and tube heat exchanger, the length of pipe required in this arrangement is impractical. Since in most of the cases available space is a major constraint, shell and tube heat exchanger may be preferred.

Special Topic: The Human Cardiovascular System as a Counter-Current Heat Exchanger

11-138C The cardiovascular counter-current mechanism is a method for blood returning to the human body core to be warmed back to core body temperature after losing heat near the skin. An artery with warmer blood is paired with a vein with cooler blood. There is significant heat transfer from the artery to the vein to warm the blood. Remember that at the extremity location, the blood temperature will be close to the environment temperature under "reasonable" scenarios.

11-139C There are two major differences. The first difference is that the fluid is the same throughout the "warm" side and the "cool" side. The second major difference is that the "warm" side's output is the input for the "cool" side. Therefore, the temperature differences are generally not as large as in conventional engineering heat exchangers.

11-140C Clothing would insulate the skin blood vessels from the environment. Therefore, the skin temperature may be closer to the core body temperature instead of closer to the environmental temperature. This would effectively reduce the load on the heat exchanger. Extreme environmental conditions may cause the skin blood vessels to be significantly colder or warmer than the core body temperature. If the skin is relatively cold (< 10°C), the counter-current mechanisms would have a significant load needed to warm blood back to core temperature. If the skin is very hot (>45°C), then the cardiovascular counter-current heat exchanger would act to cool venous blood, instead of warm venous blood.

11-141 For the given dimensions of artery, vein and blood vessels total thermal resistance of the cardiovascular system is to be determined.

Assumptions **1** Steady state conditions exist. **2** Heat transfer coefficients of artery and vein are constant and uniform.

Analysis The total thermal resistance of the cardiovascular system can be formulated as,

$$R = \left(\frac{1}{hA}\right)_{artery} + \left(\frac{1}{hA}\right)_{vein} + \frac{\ln(D_{vein} / D_{artery})}{2\pi kL}$$

$$R = \frac{1}{(300\ \mathrm{W/m^2\cdot K})\pi(0.0005\,\mathrm{m})(0.05\,\mathrm{m})} + \frac{1}{(190\ \mathrm{W/m^2\cdot K})\pi(0.0006\,\mathrm{m})(0.05\,\mathrm{m})} + \frac{\ln(600/500)}{2\pi(0.670\ \mathrm{W/m\cdot K})(0.05\,\mathrm{m})} = \mathbf{99\ K/W}$$

11-142 Reconsider Prob. 11-141 but with fouling resistance due to physiological inhomogeneities for both artery and vein.

Assumption **1** Steady state conditions exist. **2** Heat transfer coefficients of artery and vein are constant and uniform.

Analysis The total thermal resistance of the cardiovascular system can be formulated as,

$$R = \left(\frac{1}{hA}\right)_{artery} + \left(\frac{1}{hA}\right)_{vein} + \frac{\ln(D_{vein}/D_{artery})}{2\pi kL} + \left(\frac{R_f}{A}\right)_{artery} + \left(\frac{R_f}{A}\right)_{vein}$$

$$R = \frac{1}{(300\ \mathrm{W/m^2\cdot K})\pi\,(0.0005\,\mathrm{m})\,(0.05\,\mathrm{m})} + \frac{1}{(190\ \mathrm{W/m^2\cdot K})\pi\,(0.0006\,\mathrm{m})\,(0.05\,\mathrm{m})} + \frac{\ln(600/500)}{2\pi\,(0.670\ \mathrm{W/m\cdot K})(0.05\,\mathrm{m})}$$

$$+ \frac{0.0005\,\mathrm{m^2\cdot K/W}}{\pi\,(0.0005\,\mathrm{m})(0.05\,\mathrm{m})} + \frac{0.0003\,\mathrm{m^2\cdot K/W}}{\pi\,(0.0006\,\mathrm{m})(0.05\,\mathrm{m})} = \mathbf{108.7\ K/W}$$

11-143 The cardiovascular system used as a counter-current heat exchanger is used to warm venous blood. For the known temperature and mass flow rate of arterial blood, the overall blood vessel length is to be determined.

Assumption **1** Steady operating conditions. **2** Fluid properties remain constant **3** Negligible changes in the kinetic and potential energies.

Analysis The heat capacity rate of the hot and cold side of the heat exchanger is calculated as

$$C_h = \dot{m}_h c_{ph} = (0.005\,\text{kg/s})(3475\ \text{J/kg}\cdot\text{K}) = 17.38\,\text{W/K} \rightarrow C_{\max}$$
$$C_c = \dot{m}_c c_{pc} = (0.002\,\text{kg/s})(3475\,\text{J/kg}\cdot\text{K}) = 6.95\,\text{W/K} \rightarrow C_{\min}$$

Thus the capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{6.95\,\text{W/K}}{17.38\ \text{W/K}} = 0.4$$

Now, the theoretical maximum heat transfer rate is calculated as

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in})$$
$$\therefore \dot{Q}_{\max} = (6.95\,\text{W/K})(37-28)°\text{C} = 62.55\,\text{W}$$

The actual heat transfer rate is,

$$\dot{Q}_{\max} = C_c(T_{c,out} - T_{c,in})$$
$$\therefore \dot{Q}_{\max} = (6.95\,\text{W/K})(35-28)^{\text{o}}\text{C} = 48.65\,\text{W}$$

From the definition of the effectiveness of the heat exchanger we get

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{48.65\,\text{W}}{62.55\,\text{W}} = 0.778$$

Now the number of transfer units can be obtained from the ε-NTU correlations in Table 11-5.

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon-1}{c\varepsilon-1}\right) = \frac{1}{0.4-1}\ln\left(\frac{0.778-1}{0.4\times 0.778-1}\right) = 1.89$$

From the definition of NTU, we find the surface area of the heat exchanger as

$$A_s = \frac{NTU\,C_{\min}}{U} = \frac{1.89(6.95\,\text{W/K})}{125\,\text{W/m}^2\cdot\text{K}} = 0.105\,\text{m}^2$$

Hence, the length of the heat exchanger is

$$L = \frac{A_s}{\pi D} = \frac{0.105\ \text{m}^2}{\pi\,(0.05\text{m})} = \mathbf{0.668\,m}$$

11-144 The cardiovascular system used as a counter-current heat exchanger is used to warm venous blood. For the known values of overall heat transfer coefficient and inlet and exit temperatures of blood, the mass flow rate of venous and arterial blood is to be determined.

Assumptions **1** Steady state conditions exist. **2** Fluid (arterial and venous blood) properties are constant.

Analysis Since we do not know the mass flow rate of hot (arterial) and cold (venous) blood, we can find the heat transfer rate from,

$$\dot{Q} = UA_s \Delta T_{lm}$$

The log mean temperature difference for a counter-flow heat exchanger is determined as follows.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

$$\Delta T_1 = T_{h,in} - T_{c,out} = 37 - 34 = 3^o C$$

and $$\Delta T_2 = T_{h,out} - T_{c,in} = 27 - 25 = 2^o C$$

$$\Delta T_{lm} = \frac{3-2}{\ln(3/2)} = 2.47^o C$$

Thus the actual rate of heat transfer is

$$\dot{Q} = UA_s \Delta T_{lm} = (100\ \mathrm{W/m^2 \cdot K})(0.15\,\mathrm{m}^2)(2.47\ ^\circ\mathrm{C}) = 37.1\ \mathrm{W}$$

Now, the mass flow rate of hot (arterial) and cold (venous) blood can be determined from energy balance.

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in})$$

The mass flow rate of hot (arterial) blood is

$$\dot{m}_h = \frac{\dot{Q}}{c_{ph}(T_{h,in} - T_{h,out})} = \frac{37.1\mathrm{W}}{(3475\ \mathrm{J/kg \cdot K})\,(37-27)^o C} = \mathbf{1.06\,g/s}$$

The mass flow rate of cold (venous) blood is

$$\dot{m}_c = \frac{\dot{Q}}{c_{pc}(T_{cout} - T_{c,in})} = \frac{37.1\mathrm{W}}{(3475\ \mathrm{J/kg \cdot K})\,(34-25)^o C} = \mathbf{1.18\,g/s}$$

Review Problems

11-145 The inlet and outlet temperatures of the cold and hot fluids in a double-pipe heat exchanger are given. It is to be determined whether this is a parallel-flow or counter-flow heat exchanger.

Analysis In parallel-flow heat exchangers, the temperature of the cold water can never exceed that of the hot fluid. In this case $T_{\text{cold out}}$ = 50°C which is greater than $T_{\text{hot out}}$ = 45°C. Therefore this must be a counter-flow heat exchanger.

11-146 It is to be shown that when $\Delta T_1 = \Delta T_2$ for a heat exchanger, the ΔT_{lm} relation reduces to $\Delta T_{\text{lm}} = \Delta T_1 = \Delta T_2$.

Analysis When $\Delta T_1 = \Delta T_2$, we obtain

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{0}{0}$$

This case can be handled by applying L'Hospital's rule (taking derivatives of nominator and denominator separately with respect to ΔT_1 or ΔT_2). That is,

$$\Delta T_{lm} = \frac{d(\Delta T_1 - \Delta T_2)/d\Delta T_1}{d[\ln(\Delta T_1 / \Delta T_2)]/d\Delta T_1} = \frac{1}{1/\Delta T_1} = \Delta T_1 = \Delta T_2$$

11-147E Water is heated by solar-heated hot air in a double-pipe counter-flow heat exchanger. The required length of the tube is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the water and air are given to be 1.0 and 0.24 Btu/lbm.°F, respectively.

Analysis The rate of heat transfer in this heat exchanger is

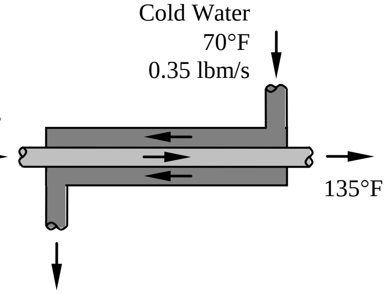
$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = (0.7\text{ lbm/s})(0.24\text{ Btu/lbm.°F})(190°\text{F} - 135°\text{F}) = 9.24\text{ Btu/s}$$

The outlet temperature of the cold water is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 70°\text{F} + \frac{9.24\text{ Btu/s}}{(0.35\text{ lbm/s})(1.0\text{ Btu/lbm.°F})} = 96.4°\text{F}$$

The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 190°\text{F} - 96.4°\text{F} = 93.6°\text{F}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 135°\text{F} - 70°\text{F} = 65°\text{F}$$



The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{93.6 - 65}{\ln(93.6/65)} = 78.43°\text{F}$$

The heat transfer surface area on the outer side of the tube is determined from

$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{9.24\text{ Btu/s}}{(20/3600\text{ Btu/s.ft}^2\text{.°F})(78.43°\text{F})} = 21.21\text{ ft}^2$$

Then the length of the tube required becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{21.21\text{ ft}^2}{\pi(0.5/12\text{ ft})} = \mathbf{162.0\ ft}$$

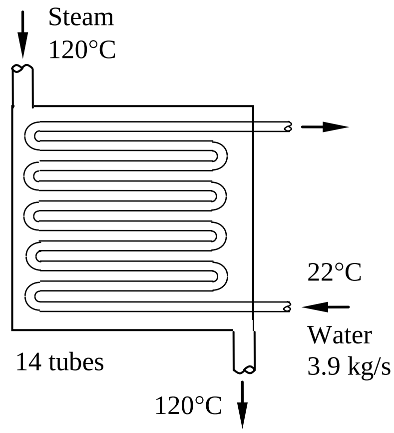
11-148 A shell-and-tube heat exchanger is used to heat water with geothermal steam condensing. The rate of heat transfer, the rate of condensation of steam, and the overall heat transfer coefficient are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The heat of vaporization of geothermal water at 120°C is given to be h_{fg} = 2203 kJ/kg and specific heat of water is given to be c_p = 4180 J/kg.°C.

Analysis (*a*) The outlet temperature of the water is

$$T_{\text{c,out}} = T_{\text{h,out}} - 46 = 120°\text{C} - 46°\text{C} = 74°\text{C}$$



Then the rate of heat transfer becomes

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \\ &= (3.9\text{ kg/s})(4.18\text{ kJ/kg.°C})(74°\text{C} - 22°\text{C}) \\ &= \mathbf{847.7\,kW}\end{aligned}$$

(*b*) The rate of condensation of steam is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\substack{\text{geothermal}\\ \text{steam}}}$$

$$847.7\text{ kW} = \dot{m}(2203\text{ kJ/kg}) \longrightarrow \dot{m} = \mathbf{0.3848\,kg/s}$$

(*c*) The heat transfer area is

$$A_i = n\pi D_i L = 14\pi(0.024\text{ m})(3.2\text{ m}) = 3.378\text{ m}^2$$

The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{\text{h,in}} - T_{\text{c,out}} = 120°\text{C} - 74°\text{C} = 46°\text{C}$$

$$\Delta T_2 = T_{\text{h,out}} - T_{\text{c,in}} = 120°\text{C} - 22°\text{C} = 98°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{46 - 98}{\ln(46/98)} = 68.75°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{74 - 22}{120 - 22} = 0.53 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{120 - 120}{74 - 22} = 0 \end{aligned}\right\} F = 1$$

Then the overall heat transfer coefficient is determined to be

$$\dot{Q} = U_i A_i F \Delta T_{\text{lm,CF}} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{\text{lm,CF}}} = \frac{847{,}700\text{ W}}{(3.378\text{ m}^2)(1)(68.75°\text{C})} = \mathbf{3650\,W/m^2.°C}$$

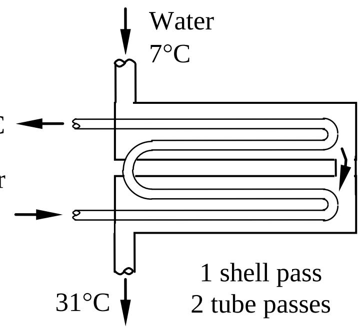
11-149 Hot water is cooled by cold water in a 1-shell pass and 2-tube passes heat exchanger. The mass flow rates of both fluid streams are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

Properties The specific heats of both cold and hot water streams are taken to be 4.18 kJ/kg.°C.

Analysis The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 60°\text{C} - 31°\text{C} = 29°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 36°\text{C} - 7°\text{C} = 29°\text{C}$$



Since $\Delta T_1 = \Delta T_2$, we have $\Delta T_{lm,CF} = 29°\text{C}$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{31-60}{7-60} = 0.45 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{7-31}{36-60} = 1.0 \end{aligned}\right\} F = 0.88 \ \text{(Fig. 11-19)}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (950 \text{ W/m}^2.°\text{C})(15 \text{ m}^2)(0.88)(29°\text{C}) = 3.64\times 10^5 \text{ W} = 364 \text{ kW}$$

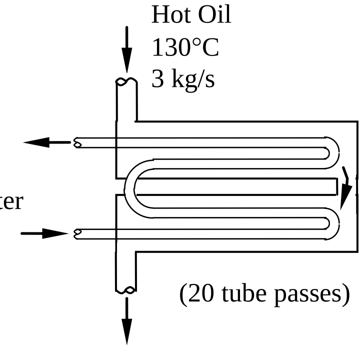
The mass flow rates of fluid streams are

$$\dot{m}_c = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{364 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(60°\text{C} - 36°\text{C})} = \mathbf{3.63 kg/s}$$

$$\dot{m}_h = \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{364 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(31°\text{C} - 7°\text{C})} = \mathbf{3.63 kg/s}$$

11-150 Water is heated by hot oil in a multi-pass shell-and-tube heat exchanger. The rate of heat transfer and the heat transfer surface area on the outer side of the tube are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.



Properties The specific heats of the water and oil are given to be 4.18 and 2.2 kJ/kg.°C, respectively.

Analysis (*a*)The rate of heat transfer in this heat exchanger is

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h,in} - T_{h,out}) = (3\,\text{kg/s})(2.2\,\text{kJ/kg.°C})(130°\text{C} - 60°\text{C}) = \mathbf{462\,kW}$$

(*b*) The outlet temperature of the cold water is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c c_{pc}} = 20°\text{C} + \frac{462\,\text{kW}}{(3\,\text{kg/s})(4.18\,\text{kJ/kg.°C})} = 56.84°\text{C}$$

The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 130°\text{C} - 56.84°\text{C} = 73.16°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 60°\text{C} - 20°\text{C} = 40°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{73.16 - 45}{\ln(73.16/45)} = 57.94°\text{C}$$

and

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{56.84 - 20}{130 - 20} = 0.33 \\ R &= \frac{T_2 - T_1}{t_2 - t_1} = \frac{130 - 60}{56.84 - 20} = 1.90 \end{aligned}\right\} F = 0.97$$

The heat transfer surface area on the outer side of the tube is then determined from

$$\dot{Q} = UA_s F\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{UF\Delta T_{lm}} = \frac{462\,\text{kW}}{(0.22\,\text{kW/m}^2.°\text{C})(0.97)(57.94°\text{C})} = \mathbf{37.4\,m^2}$$

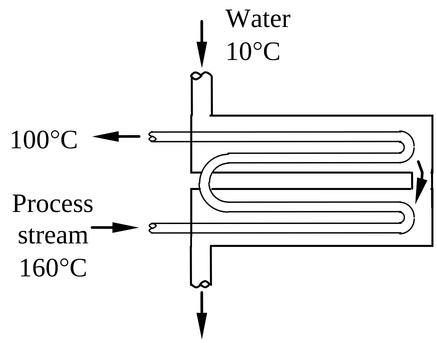
11-151 Water is used to cool a process stream in a shell and tube heat exchanger. The tube length is to be determined for one tube pass and four tube pass cases.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The properties of process stream and water are given in problem statement.

Analysis (a) The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (47\ \text{kg/s})(3.5\ \text{kJ/kg}\cdot°\text{C})(160 - 100)°\text{C} = 9870\ \text{kW}$$



The outlet temperature of water is determined from

$$\dot{Q} = \dot{m}_c c_c (T_{c,out} - T_{c,in})$$

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}_c C_c} = 10°\text{C} + \frac{9870\ \text{kW}}{(66\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})} = 45.8°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 160°\text{C} - 45.8°\text{C} = 114.2°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 100°\text{C} - 10°\text{C} = 90°\text{C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{114.2 - 90}{\ln\left(\frac{114.2}{90}\right)} = 101.6°\text{C}$$

The Reynolds number is

$$V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{N_{tube}\rho\pi D^2/4} = \frac{(47\ \text{kg/s})}{(100)(950\ \text{kg/m}^3)\pi(0.025\ \text{m})^2/4} = 1.008\ \text{m/s}$$

$$\text{Re} = \frac{VD\rho}{\mu} = \frac{(1.008\ \text{m/s})(0.025\ \text{m})(950\ \text{kg/m}^3)}{0.002\ \text{kg/m}\cdot\text{s}} = 11{,}968$$

which is greater than 10,000. Therefore, we have turbulent flow. We assume fully developed flow and evaluate the Nusselt number from

$$\text{Pr} = \frac{\mu c_p}{k} = \frac{(0.002\ \text{kg/m}\cdot\text{s})(3500\ \text{J/kg}\cdot°\text{C})}{0.50\ \text{W/m}\cdot°\text{C}} = 14$$

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(11{,}968)^{0.8}(14)^{0.3} = 92.9$$

Heat transfer coefficient on the inner surface of the tubes is

$$h_i = \frac{k}{D}Nu = \frac{0.50\ \text{W/m.°C}}{0.025\ \text{m}}(92.9) = 1858\ \text{W/m}^2.°\text{C}$$

Disregarding the thermal resistance of the tube wall the overall heat transfer coefficient is determined from

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{1858} + \frac{1}{4000}} = 1269\ \text{W/m}^2\cdot°\text{C}$$

The correction factor for one shell pass and one tube pass heat exchanger is $F = 1$. The tube length is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$9870\ \text{kW} = (1.269\ \text{kW/m}^2\cdot\text{C})[100\pi(0.025\ \text{m})L](1)(101.6°\text{C})$$

$$L = \mathbf{9.75\ m}$$

(b) For 1 shell pass and 4 tube passes, there are 100/4=25 tubes per pass and this will increase the velocity fourfold. We repeat the calculations for this case as follows:

$$V = 4\times 1.008 = 4.032 \text{ m/s}$$

$$\text{Re} = 4\times 11{,}968 = 47{,}872$$

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(47{,}872)^{0.8}(14)^{0.3} = 281.6$$

$$h_i = \frac{k}{D}Nu = \frac{0.50 \text{ W/m.°C}}{0.025 \text{ m}}(281.6) = 5632 \text{ W/m}^2\text{.°C}$$

$$U = \frac{1}{\frac{1}{h_i}+\frac{1}{h_o}} = \frac{1}{\frac{1}{5632}+\frac{1}{4000}} = 2339 \text{ W/m}^2\cdot\text{°C}$$

The correction factor is determined from Fig. 11-19:

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{100-160}{10-160} = 0.4 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{10-45.8}{100-160} = 0.60 \end{aligned}\right\} F = 0.96$$

The tube length is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$

$$9870 \text{ kW} = (2.339 \text{ kW/m}^2\cdot\text{C})\left[100\pi(0.025 \text{ m})L\right](0.96)(101.6\text{°C})$$

$$L = \mathbf{5.51m}$$

11-152 A hydrocarbon stream is heated by a water stream in a 2-shell passes and 4-tube passes heat exchanger. The rate of heat transfer and the mass flow rates of both fluid streams and the fouling factor after usage are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heat of HC is given to be 2 kJ/kg.°C. The specific heat of water is taken to be 4.18 kJ/kg.°C.

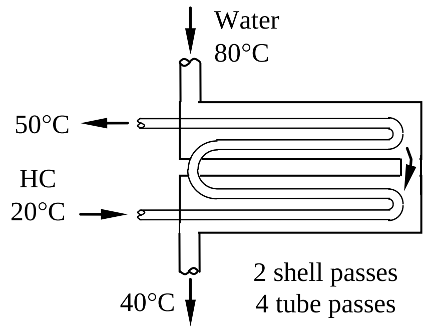
Analysis (a) The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 50°\text{C} = 30°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 20°\text{C} = 20°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{30-20}{\ln(30/20)} = 24.66°\text{C}$$

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{50-20}{80-20} = 0.5 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{80-40}{50-20} = 1.33 \end{aligned}\right\} F = 0.90 \ \ \text{(Fig. 11-19)}$$



The overall heat transfer coefficient of the heat exchanger is

$$U = \frac{1}{\dfrac{1}{h_i} + \dfrac{1}{h_o}} = \frac{1}{\dfrac{1}{1600} + \dfrac{1}{2500}} = 975.6 \text{ W/m}^2 \cdot °\text{C}$$

The rate of heat transfer in this heat exchanger is

$$\dot{Q} = UA_s F \Delta T_{lm,CF} = (975.6 \text{ W/m}^2.°\text{C})[160\pi(0.02 \text{ m})(1.5 \text{ m})](0.90)(24.66°\text{C}) = 3.265\times10^5 \text{ W} = \mathbf{326.5 kW}$$

The mass flow rates of fluid streams are

$$\dot{m}_c = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{326.5 \text{ kW}}{(2.0 \text{ kJ/kg.°C})(50°\text{C} - 20°\text{C})} = \mathbf{5.44 kg/s}$$

$$\dot{m}_h = \frac{\dot{Q}}{c_p(T_{in} - T_{out})} = \frac{326.5 \text{ kW}}{(4.18 \text{ kJ/kg.°C})(80°\text{C} - 40°\text{C})} = \mathbf{1.95 kg/s}$$

(b) The rate of heat transfer in this case is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_\text{c} = (5.44 \text{ kg/s})(2.0 \text{ kJ/kg.°C})(45°\text{C} - 20°\text{C}) = 272 \text{ kW}$$

This corresponds to a 17% decrease in heat transfer. The outlet temperature of the hot fluid is

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_\text{h}$$

$$272 \text{ kW} = (1.95 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(80°\text{C} - T_{h,out})$$

$$T_{h,out} = 46.6°\text{C}$$

The logarithmic temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 80°\text{C} - 45°\text{C} = 35°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 46.6°\text{C} - 20°\text{C} = 26.6°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{35-26.6}{\ln(35/26.6)} = 30.61°\text{C}$$

$$\left.\begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{45 - 20}{80 - 20} = 0.42 \\ R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{80 - 46.6}{45 - 20} = 1.34 \end{array}\right\} F = 0.97 \text{ (Fig. 11-19)}$$

The overall heat transfer coefficient is

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$
$$272{,}000 \text{ W} = U\left[160\pi(0.02 \text{ m})(1.5 \text{ m})\right](0.97)(30.61°\text{C})$$
$$U = 607.5 \text{ W/m}^2.°\text{C}$$

The fouling factor is determined from

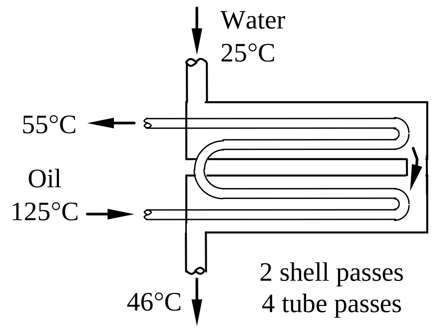
$$R_f = \frac{1}{U_{dirty}} - \frac{1}{U_{clean}} = \frac{1}{607.5} - \frac{1}{975.6} = \mathbf{6.21 \times 10^{-4} \ m^2 \cdot °C/W}$$

11-153 Oil is cooled by water in a 2-shell passes and 4-tube passes heat exchanger. The mass flow rate of water and the surface area are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

Properties The specific heat of oil is given to be 2 kJ/kg.°C. The specific heat of water is taken to be 4.18 kJ/kg.°C.

Analysis The logarithmic mean temperature difference for counter-flow arrangement and the correction factor F are



$$\Delta T_1 = T_{h,in} - T_{c,out} = 125°\text{C} - 46°\text{C} = 79°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 55°\text{C} - 25°\text{C} = 30°\text{C}$$

$$\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{79-30}{\ln(79/30)} = 50.61°\text{C}$$

$$\left.\begin{array}{l} P = \dfrac{t_2 - t_1}{T_1 - t_1} = \dfrac{55-125}{25-125} = 0.7 \\[2ex] R = \dfrac{T_1 - T_2}{t_2 - t_1} = \dfrac{25-46}{55-125} = 0.3 \end{array}\right\} F = 0.97 \ \text{(Fig. 11-19)}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}_h c_h (T_{h,in} - T_{h,out}) = (10\ \text{kg/s})(2.0\ \text{kJ/kg}\cdot°\text{C})(125-55)°\text{C} = 1400\ \text{kW}$$

The mass flow rate of water is

$$\dot{m}_w = \frac{\dot{Q}}{c_p (T_{out} - T_{in})} = \frac{1400\ \text{kW}}{(4.18\ \text{kJ/kg.°C})(46°\text{C} - 25°\text{C})} = \mathbf{15.9\ kg/s}$$

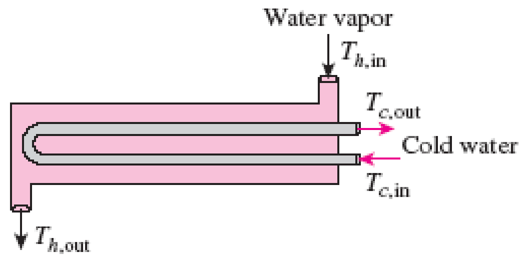
The surface area of the heat exchanger is determined to be

$$\dot{Q} = UAF\Delta T_{lm}$$
$$1400\ \text{kW} = (0.9\ \text{kW/m}^2\cdot\text{C})A_s(0.97)(50.61°\text{C})$$
$$A_s = \mathbf{31.7\ m^2}$$

11-154 Saturated water vapor condenses in a 1-shell and 2-tube heat exchanger, the outlet temperature of the cold water and the heat transfer rate for the heat exchanger are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the cold water is given to be c_{pc} = 4179 J/kg·K.



Analysis The log mean temperature difference for the counter-flow arrangement is

$$\Delta T_{\text{lm, CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(100 - T_{c,\text{out}}) - (100 - 15)}{\ln[(100 - T_{c,\text{out}})/(100 - 15)]}$$

The heat transfer rate can be written as

$$\dot{Q} = UA_s F \Delta T_{\text{lm, CF}} = (2000\ \text{W/m}^2 \cdot \text{K})(0.5\ \text{m}^2)\frac{(100 - T_{c,\text{out}}) - (100 - 15)}{\ln[(100 - T_{c,\text{out}})/(100 - 15)]}\ \text{K} \qquad (1)$$

where F = 1 for condensation process. From energy balance, the heat transfer rate can also be written as

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,\text{out}} - T_{c,\text{in}}) = (0.5\ \text{kg/s})(4179\ \text{J/kg} \cdot \text{K})(T_{c,\text{out}} - 15)\ \text{K} \qquad (2)$$

The outlet temperature of the cold water and the heat transfer rate can be determined by solving Eqs. (1) and (2) simultaneously. Copy the following lines and paste on a blank EES screen:

```
Q_dot=(2000)*(0.5)*((100-T_co)-(100-15))/ln((100-T_co)/(100-15))
Q_dot=(0.5)*(4179)*(T_co-15)
```

Solving by EES software, we get

$$\dot{Q} = \mathbf{67600\ W}$$

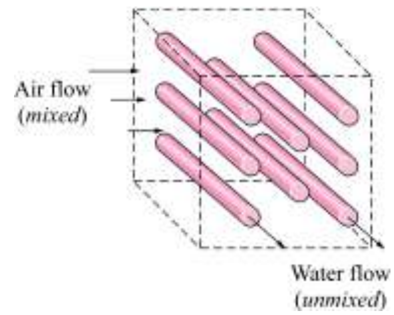
$$T_{c,\text{out}} = \mathbf{47.3°C}$$

Discussion The value of the correction factor is F = 1 for process involving phase-change (boiling or condensation).

11-155 A single-pass cross-flow heat exchanger uses hot air (mixed) to heat water (unmixed), and the required surface area of the heat exchanger is to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of water at the average temperature of 55°C is c_{pc} = 4183 J/kg·K (Table A-9); the specific heat of air at the average temperature of 160°C is c_{ph} = 1016 J/kg·K (Table A-15).



Analysis Using Fig. 11-19*d*, the correction factor can be determined to be

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{80-30}{220-30} = 0.26 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{220-100}{80-30} = 2.4 \end{aligned}\right\} F \approx 0.92 \qquad \text{(Fig. 11-19}d\text{)}$$

Using energy balance on the cold fluid, we have

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c,\text{out}} - T_{c,\text{in}}) = (3\ \text{kg/s})(4183\ \text{J/kg}\cdot\text{K})(80-30)\ \text{K} = 6.275\times10^5\ \text{W}$$

The log mean temperature difference for the counter-flow arrangement is

$$\Delta T_{\text{lm, CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{(220-80)-(100-30)}{\ln[(220-80)/(100-30)]}\ °\text{C} = 101°\text{C}$$

Thus, the surface area can be determined using

$$\dot{Q} = UA_s F\Delta T_{\text{lm, CF}} \qquad \rightarrow \qquad A_s = \frac{\dot{Q}}{UF\Delta T_{\text{lm, CF}}}$$

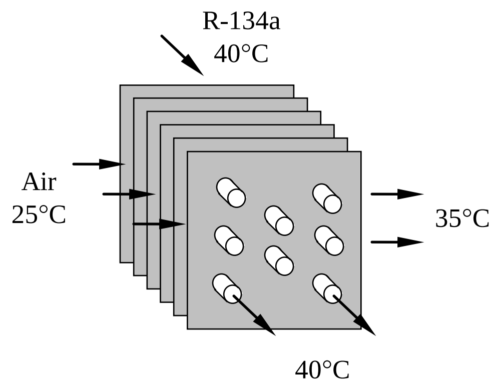
$$A_s = \frac{6.275\times10^5\ \text{W}}{(200\ \text{W/m}^2\cdot\text{K})(0.92)(101\ \text{K})} = \mathbf{33.7\ m^2}$$

Discussion If there is fouling, it will reduce the rate of heat transfer of the heat exchanger.

11-156 Refrigerant-134a is condensed by air in the condenser of a room air conditioner. The heat transfer area on the refrigerant side is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heat of air is given to be 1.005 kJ/kg.°C.



Analysis The temperature differences at the two ends are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 40°\text{C} - 35°\text{C} = 5°\text{C}$$
$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°\text{C} - 25°\text{C} = 15°\text{C}$$

The logarithmic mean temperature difference is

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{5-15}{\ln(5/15)} = 9.102°\text{C}$$

The heat transfer surface area on the outer side of the tube is determined from

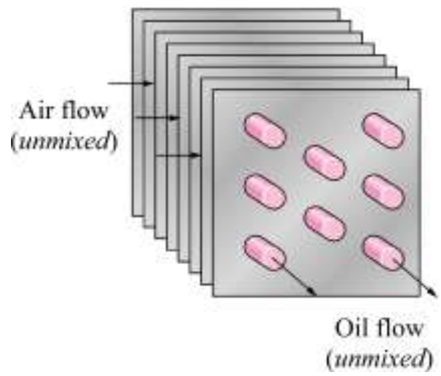
$$\dot{Q} = UA_s\Delta T_{lm} \longrightarrow A_s = \frac{\dot{Q}}{U\Delta T_{lm}} = \frac{(15{,}000/3600)\text{ kW}}{(0.150\text{ kW/m}^2.°\text{C})(9.102°\text{C})} = \mathbf{3.05\ m^2}$$

11-157 Oil in an engine is being cooled by air in a cross-flow heat exchanger, where both fluids are unmixed; with a specified correction factor, the outlet temperatures of the oil and air are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heats of oil and air are given to be c_{ph} = 2047 J/kg·K and c_{pc} = 1007 J/kg·K, respectively.

Analysis On the shell side (air),



$$(\dot{m}_c c_{pc})_{\text{shellside}} = (0.21\,\text{kg/s})(1007\,\text{J/kg}\cdot\text{K}) = 211.5\,\text{W/K}$$

On the tube side (oil),

$$(\dot{m}_h c_{ph})_{\text{tubeside}} = (0.026\,\text{kg/s})(2047\,\text{J/kg}\cdot\text{K}) = 53.22\,\text{W/K}$$

Then, we have

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}_h c_{ph})_{\text{tubeside}}}{(\dot{m}_c c_{pc})_{\text{shellside}}} = \frac{53.22\,\text{W/K}}{211.5\,\text{W/K}} = 0.2516$$

With R = 0.25 and F = 0.96, using Fig. 11-19*c* yields

$$P = \frac{t_2 - t_1}{T_1 - t_1} \approx 0.60$$

The outlet temperature of the oil is

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad \rightarrow \quad T_{h,\text{out}} = t_2 = P(T_1 - t_1) + t_1 = (0.6)(30 - 75)°\text{C} + 75°\text{C} = \mathbf{48.0°C}$$

The outlet temperature of the air is

$$R = \frac{T_1 - T_2}{t_2 - t_1} \quad \rightarrow \quad T_{c,\text{out}} = T_2 = T_1 - R(t_2 - t_1) = 30°\text{C} - (0.2516)(48 - 75)°\text{C} = \mathbf{36.8°C}$$

Discussion The outlet temperatures can be determined using the effectiveness-NTU method without knowing the value of the correction factor (F).

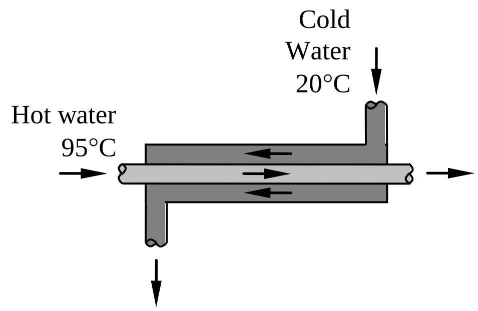
11-158 A water-to-water counter-flow heat exchanger is considered. The outlet temperature of the cold water, the effectiveness of the heat exchanger, the mass flow rate of the cold water, and the heat transfer rate are to be determined.

.***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of both the cold and the hot water are given to be 4.18 kJ/kg.°C.

Analysis (*a*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = 1.5\dot{m}_c(4.18\,\text{kJ/kg.°C}) = 6.27\dot{m}_c$$
$$C_c = \dot{m}_c c_{pc} = \dot{m}_c(4.18\,\text{kJ/kg.°C}) = 4.18\dot{m}_c$$



Therefore,

$$C_{\min} = C_c = 4.18\dot{m}_c$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{4.18\dot{m}_c}{6.27\dot{m}_c} = 0.6667$$

The rate of heat transfer can be expressed as

$$\dot{Q} = C_c(T_{\text{c,out}} - T_{\text{c,in}}) = (4.18\dot{m}_c)(T_{\text{c,out}} - 20)$$
$$\dot{Q} = C_h(T_{\text{h,in}} - T_{\text{h,out}}) = (6.27\dot{m}_c)\left[95 - (T_{\text{c,out}} + 15)\right] = (6.27\dot{m}_c)(80 - T_{\text{c,out}})$$

Setting the above two equations equal to each other we obtain the outlet temperature of the cold water

$$\dot{Q} = 4.18\dot{m}_c(T_{\text{c,out}} - 20) = 6.27\dot{m}_c(80 - T_{\text{c,out}})$$
$$4.18(T_{\text{c,out}} - 20) = 6.27(80 - T_{\text{c,out}}) \longrightarrow T_{\text{c,out}} = \mathbf{56.0°C}$$

(*b*) The effectiveness of the heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{\text{c,out}} - T_{\text{c,in}})}{C_c(T_{\text{h,in}} - T_{\text{c,in}})} = \frac{4.18\dot{m}_c(56.0 - 20)}{4.18\dot{m}_c(95 - 20)} = \mathbf{0.480}$$

(*c*) The NTU of this heat exchanger is determined from

$$NTU = \frac{1}{c-1}\ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right) = \frac{1}{0.6667 - 1}\ln\left(\frac{0.480 - 1}{0.480 \times 0.6667 - 1}\right) = 0.8048$$

Then, from the definition of NTU, we obtain the mass flow rate of the cold fluid:

$$NTU = \frac{UA_s}{C_{\min}} \longrightarrow 0.8048 = \frac{1.400\,\text{kW/°C}}{4.18\dot{m}_c} \longrightarrow \dot{m}_c = \mathbf{0.4162\,kg/s}$$

(*d*) The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc}(T_{\text{c,out}} - T_{\text{c,in}}) = (0.4162\,\text{kg/s})(4.18\,\text{kJ/kg.°C})(56.0 - 20)\text{°C} = \mathbf{62.6\,kW}$$

11-159 Water is heated by geothermal water in a double-pipe counter-flow heat exchanger. The mass flow rate of the geothermal water and the outlet temperatures of both fluids are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

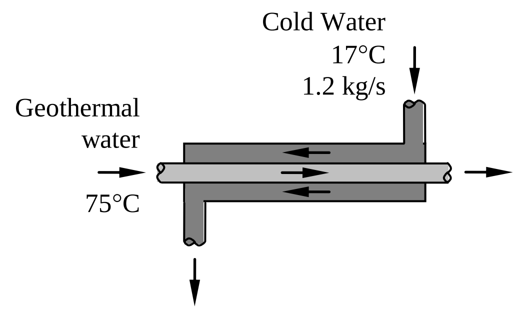
Properties The specific heats of the geothermal water and the cold water are given to be 4.25 and 4.18 kJ/kg.°C, respectively.

Analysis The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = \dot{m}_h (4.25\text{ kJ/kg.°C}) = 4.25\dot{m}_h$$

$$C_c = \dot{m}_c c_{pc} = (1.2\text{ kg/s})(4.18\text{ kJ/kg.°C}) = 5.016\text{ kW/°C}$$

$$C_{\min} = C_c = 5.016\text{ kW/°C}$$



and $$c = \frac{C_{\min}}{C_{\max}} = \frac{5.016}{4.25\dot{m}_h} = \frac{1.1802}{\dot{m}_h}$$

The NTU of this heat exchanger is

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.480\text{ kW/m}^2.\text{°C})(25\text{ m}^2)}{5.016\text{ kW/°C}} = 2.392$$

Using the effectiveness relation, we find the capacity ratio

$$\varepsilon = \frac{1-\exp[-\text{NTU}(1-c)]}{1-c\exp[-\text{NTU}(1-c)]} \longrightarrow 0.823 = \frac{1-\exp[-2.392(1-c)]}{1-c\exp[-2.392(1-c)]} \longrightarrow c = 0.494$$

Then the mass flow rate of geothermal water is determined from

$$c = \frac{1.1802}{\dot{m}_h} \longrightarrow 0.494 = \frac{1.1802}{\dot{m}_h} \longrightarrow \dot{m}_h = \mathbf{2.39\,kg/s}$$

The maximum heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{\text{h,in}} - T_{\text{c,in}}) = (5.016\text{ kW/°C})(75\text{°C} - 17\text{°C}) = 290.9\text{ kW}$$

Then the actual rate of heat transfer rate becomes

$$\dot{Q} = \varepsilon\dot{Q}_{\max} = (0.823)(290.9\text{ kW}) = 239.4\text{ kW}$$

The outlet temperatures of the geothermal and cold waters are determined to be

$$\dot{Q} = C_c(T_{\text{c,out}} - T_{\text{c,in}}) \longrightarrow 239.4\text{ kW} = (5.016\text{ kW/°C})(T_{\text{c,out}} - 17) \longrightarrow T_{\text{c,out}} = \mathbf{64.7°C}$$

$$\dot{Q} = \dot{m}_h c_{ph}(T_{\text{h,in}} - T_{\text{h,out}})$$

$$239.4\text{ kW} = (2.39\text{ kg/s})(4.25\text{ kJ/kg.°C})(75 - T_{\text{h,out}}) \longrightarrow T_{\text{h,out}} = \mathbf{51.4°C}$$

11-160 A cross-flow heat exchanger with both fluids unmixed has a specified overall heat transfer coefficient, (*a*) the exit temperature of the hot fluid and (*b*) the rate of heat transfer in the heat exchanger are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Analysis (*a*) The heat capacity rates are given as

$$C_h = C_{\min} = 40{,}000 \text{ W/K} \qquad \text{and} \qquad C_c = C_{\max} = 80{,}000 \text{ W/K}$$

The capacity ratio is

$$c = \frac{C_{\min}}{C_{\max}} = \frac{C_h}{C_c} = \frac{40{,}000 \text{ W/K}}{80{,}000 \text{ W/K}} = 0.5$$

The NTU of the heat exchanger is

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(200 \text{ W/m}^2\cdot\text{K})(400 \text{ m}^2)}{40{,}000 \text{ W/K}} = 2.0$$

Using the equation listed in Table 11-4, the heat transfer effectiveness is

$$\begin{aligned}\varepsilon &= 1-\exp\left\{\frac{\text{NTU}^{0.22}}{c}[\exp(-c\,\text{NTU}^{0.78})-1]\right\}\\ &= 1-\exp\left(\frac{2.0^{0.22}}{0.5}\{\exp[-(0.5)(2.0)^{0.78}]-1\}\right)\\ &= 0.7388\end{aligned}$$

From the definition of heat transfer effectiveness,

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h(T_{h,\text{in}}-T_{h,\text{out}})}{C_{\min}(T_{h,\text{in}}-T_{c,\text{in}})} = \frac{C_h(T_{h,\text{in}}-T_{h,\text{out}})}{C_h(T_{h,\text{in}}-T_{c,\text{in}})}$$

$$T_{h,\text{out}} = T_{h,\text{in}} - \varepsilon(T_{h,\text{in}}-T_{c,\text{in}}) = 80°\text{C} - (0.7388)(80°\text{C}-20°\text{C}) = \mathbf{35.7°C}$$

(*b*) The rate of heat transfer in the heat exchanger is

$$\dot{Q} = C_h(T_{h,\text{in}}-T_{h,\text{out}}) = (40{,}000 \text{ W/K})(80°\text{C}-35.7°\text{C}) = \mathbf{1.77\times10^6\ W}$$

Discussion The rate of heat transfer in the heat exchanger can also be calculated using

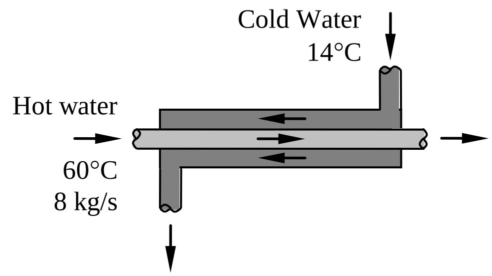
$$\dot{Q} = C_c(T_{c,\text{out}}-T_{c,\text{in}})$$

11-161 A water-to-water heat exchanger is proposed to preheat the incoming cold water by the drained hot water in a plant to save energy. The heat transfer rating of the heat exchanger and the amount of money this heat exchanger will save are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The specific heat of the hot water is given to be 4.18 kJ/kg.°C.

Analysis The maximum rate of heat transfer is



$$\begin{aligned}\dot{Q}_{\max} &= \dot{m}_h c_{ph}(T_{h,in} - T_{c,in}) \\ &= (8/60\ \text{kg/s})(4.18\ \text{kJ/kg.°C})(60°\text{C} - 14°\text{C}) \\ &= 25.6\ \text{kW}\end{aligned}$$

Noting that the heat exchanger will recover 72% of it, the actual heat transfer rate becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.72)(25.6\ \text{kJ/s}) = \mathbf{18.43 kW}$$

which is the heat transfer rating. The operating hours per year are

The annual operating hours = (8 h/day)(5 days/week)(52 week/year) = 2080 h/year

The energy saved during the entire year will be

Energy saved = (heat transfer rate)(operating time)

= (18.43 kJ/s)(2080 h/year)(3600 s/h)

= $1.38\text{x}10^8$ kJ/year

Then amount of fuel and money saved will be

$$\text{Fuel saved} = \frac{\text{Energy saved}}{\text{Furnace efficiency}} = \frac{1.38\times10^8\ \text{kJ/year}}{0.78}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = 1677\ \text{therms/year}$$

Money saved = (fuel saved)(the price of fuel)

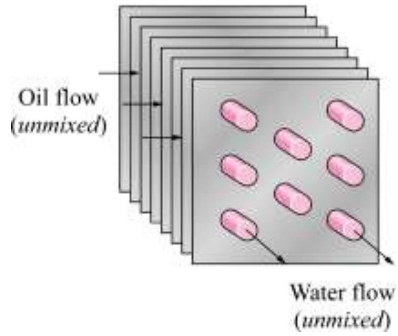
= (1677 therms/year)($1.00/therm)

= **$1677/year**

11-162 A single-pass cross-flow heat exchanger with both fluids unmixed, (*a*) the NTU value and (*b*) the value of the overall heat transfer coefficient are to be determined.

Assumptions **1** Steady operating condition exists. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible. **3** Fluid properties are constant. **4** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties The properties of oil are given to be c_{ph} = 1.93 kJ/kg·K and ρ = 870 kg/m^3.



Analysis (*a*) The mass flow rate of oil (hot fluid) is

$$\dot{m}_h = \rho\dot{\mathcal{V}} = (870\,\text{kg/m}^3)(0.19\,\text{m}^3\text{/min})(1/60\,\text{min/s}) = 2.755\,\text{kg/s}$$

The heat capacity rate for the hot fluid is

$$C_h = \dot{m}_h c_{ph} = (2.755\,\text{kg/s})(1930\,\text{J/kg}\cdot\text{K}) = 5317\,\text{W/K}$$

Using energy balance, we have

$$C_c(T_{c,\text{out}} - T_{c,\text{in}}) = C_h(T_{h,\text{in}} - T_{h,\text{out}}) \qquad \rightarrow \qquad \frac{C_c}{C_h} = \frac{T_{h,\text{in}} - T_{h,\text{out}}}{T_{c,\text{out}} - T_{c,\text{in}}} = \frac{38-29}{33-16} = 0.5294$$

or $$c = \frac{C_c}{C_h} = \frac{C_{\min}}{C_{\max}} = 0.5294$$

The heat transfer effectiveness is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_{\min}(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{C_c(T_{c,\text{out}} - T_{c,\text{in}})}{C_c(T_{h,\text{in}} - T_{c,\text{in}})} = \frac{33-16}{38-16} = 0.7727$$

From Table 11-4, the NTU value can be determined from

$$\varepsilon = 1 - \exp\left\{\frac{\text{NTU}^{0.22}}{c}[\exp(-c\,\text{NTU}^{0.78}) - 1]\right\}$$

Copy the following lines and paste on a blank EES screen to solve the above equation:

```
c=0.5294
epsilon=0.7727
epsilon=1-exp(NTU^0.22/c*(exp(-c*NTU^0.78)-1))
```

Solving by EES software, we get

$$\text{NTU} = \mathbf{2.39}$$

(*b*) The value of the overall heat transfer coefficient is

$$\text{NTU} = \frac{UA_s}{C_{\min}} \qquad \rightarrow \qquad U = \text{NTU}\frac{C_{\min}}{A_s} = \text{NTU}\frac{cC_h}{A_s}$$

$$U = (2.39)\frac{(0.5294)(5317\text{ W/K})}{20\,\text{m}^2} = \mathbf{336\ W/m^2\cdot K}$$

Discussion Using Figure 11-27*c*, the NTU value is found to be approximately NTU ≈ 2.4.

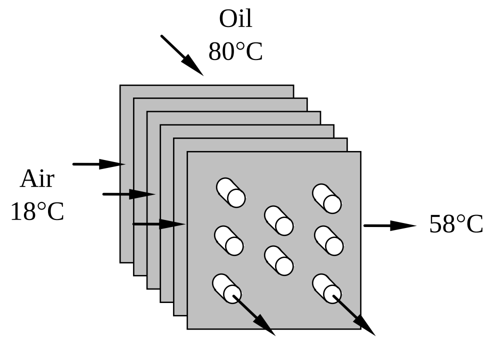
11-163 Air is to be heated by hot oil in a cross-flow heat exchanger with both fluids unmixed. The effectiveness of the heat exchanger, the mass flow rate of the cold fluid, and the rate of heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** The overall heat transfer coefficient is constant and uniform.

Properties The specific heats of the air and the oil are given to be 1.006 and 2.15 kJ/kg.°C, respectively.

Analysis (*a*) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = 0.5\dot{m}_c (2.15\,\text{kJ/kg.°C}) = 1.075\dot{m}_c$$
$$C_c = \dot{m}_c c_{pc} = \dot{m}_c (1.006\,\text{kJ/kg.°C}) = 1.006\dot{m}_c$$



Therefore,

$$C_{\min} = C_c = 1.006\dot{m}_c$$

and

$$c = \frac{C_{\min}}{C_{\max}} = \frac{1.006\dot{m}_c}{1.075\dot{m}_c} = 0.936$$

The effectiveness of the heat exchanger is determined from

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c (T_{\text{c,out}} - T_{\text{c,in}})}{C_c (T_{\text{h,in}} - T_{\text{c,in}})} = \frac{58-18}{80-18} = \mathbf{0.645}$$

(*b*) The NTU of this heat exchanger is expressed as

$$NTU = \frac{UA_s}{C_{\min}} = \frac{(0.750\,\text{kW/°C})}{1.006\dot{m}_c} = \frac{0.7455}{\dot{m}_c}$$

The NTU of this heat exchanger can also be determined from

$$NTU = -\frac{\ln\left[c\ln(1-\varepsilon)+1\right]}{c} = -\frac{\ln\left[0.936\times\ln(1-0.645)+1\right]}{0.936} = 3.724$$

Then the mass flow rate of the air is determined to be

$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow 3.724 = \frac{(0.750\,\text{kW/°C})}{1.006\dot{m}_c} \longrightarrow \dot{m}_c = \mathbf{0.20\,kg/s}$$

(*c*) The rate of heat transfer is determined from

$$\dot{Q} = \dot{m}_c c_{pc} (T_{\text{c,out}} - T_{\text{c,in}}) = (0.20\,\text{kg/s})(1.006\,\text{kJ/kg.°C})(58\text{-}18)\text{°C} = \mathbf{8.05\,kW}$$

11-164 The inlet conditions of hot and cold fluid streams in a heat exchanger are given. The outlet temperatures of both streams are to be determined using LMTD and the effectiveness-NTU methods.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of hot and cold fluid streams are given to be 2.0 and 4.2 kJ/kg.°C, respectively.

Analysis (a) The rate of heat transfer can be expressed as

$$\dot{Q} = \dot{m}c_p(T_{h,in} - T_{h,out}) = (2700/3600 \text{ kg/s})(2.0 \text{ kJ/kg.°C})(120 - T_{h,out}) = 1.5(120 - T_{h,out}) \quad (1)$$

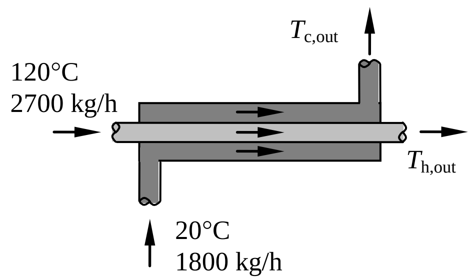
$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) = (1800/3600 \text{ kg/s})(4.2 \text{ kJ/kg.°C})(T_{c,out} - 20) = 2.1(T_{c,out} - 20) \quad (2)$$

The heat transfer can also be expressed using the logarithmic mean temperature difference as

$$\Delta T_1 = T_{h,in} - T_{c,in} = 120°\text{C} - 20°\text{C} = 100°\text{C}$$

$$\Delta T_2 = T_{h,out} - T_{c,out}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)}$$



$$\dot{Q} = UA\Delta T_{lm} = \frac{\dot{Q}_{hc,m}}{A\Delta T_{lm}}$$

$$= (2.0 \text{ kW/m}^2\cdot°\text{C})(0.50 \text{ m}^2)\frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)} = \frac{100 - (T_{h,out} - T_{c,out})}{\ln\left(\frac{100}{T_{h,out} - T_{c,out}}\right)} \quad (3)$$

Now we have three expressions for heat transfer with three unknowns: $\dot{Q}$, $T_{h,out}$, $T_{c,out}$. Solving them using an equation solver such as EES, we obtain

$$\dot{Q} = 59.6 \text{ kW}$$

$$T_{h,out} = \mathbf{80.3°C}$$

$$T_{c,out} = \mathbf{48.4°C}$$

(b) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (2700/3600 \text{ kg/s})(2.0 \text{ kJ/kg.°C}) = 1.5 \text{ kW/°C}$$

$$C_c = \dot{m}_c c_{pc} = (1800/3600 \text{ kg/s})(4.2 \text{ kJ/kg.°C}) = 2.1 \text{ kW/°C}$$

Therefore

$$C_{\min} = C_h = 1.5 \text{ kW/°C}$$

which is the smaller of the two heat capacity rates. The heat capacity ratio and the NTU are

$$c = \frac{C_{\min}}{C_{\max}} = \frac{1.5}{2.1} = 0.7143$$

$$NTU = \frac{UA}{C_{\min}} = \frac{(2.0 \text{ kW/m}^2\cdot\text{C})(0.50 \text{ m}^2)}{1.5 \text{ kW/°C}} = 0.6667$$

The effectiveness of this parallel-flow heat exchanger is

$$\varepsilon = \frac{1 - \exp[-NTU(1+c)]}{1+c} = \frac{1 - \exp[-(0.6667)(1+0.7143)]}{1+0.7143} = 0.3973$$

The maximum heat transfer rate is

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (1.5\,\text{kW/°C})(120\text{°C} - 20\text{°C}) = 150\,\text{kW}$$

The actual heat transfer rate is

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.3973)(150) = 59.60\,\text{kW}$$

Then the outlet temperatures are determined to be

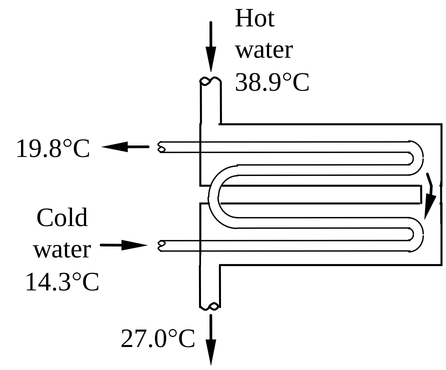
$$\dot{Q} = C_c(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{C_c} = 20\text{°C} + \frac{59.60\,\text{kW}}{2.1\,\text{kW/°C}} = \mathbf{48.4°C}$$

$$\dot{Q} = C_h(T_{h,in} - T_{h,out}) \longrightarrow T_{h,out} = T_{h,in} - \frac{\dot{Q}}{C_h} = 120\text{°C} - \frac{59.60\,\text{kW}}{1.5\,\text{kW/°C}} = \mathbf{80.3°C}$$

Discussion The results obtained by two methods are same as expected. However, the effectiveness-NTU method is easier for this type of problems.

11-165 The inlet and exit temperatures and the volume flow rates of hot and cold fluids in a heat exchanger are given. The rate of heat transfer to the cold water, the overall heat transfer coefficient, the fraction of heat loss, the heat transfer efficiency, the effectiveness, and the NTU of the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Changes in the kinetic and potential energies of fluid streams are negligible. **3** Fluid properties are constant.



Properties The densities of hot water and cold water at the average temperatures of (38.9+27.0)/2 = 33.0°C and (14.3+19.8)/2 = 17.1°C are 994.8 and 998.6 kg/m^3, respectively. The specific heat at the average temperature is 4178 J/kg.°C for hot water and 4184 J/kg.°C for cold water (Table A-9).

Analysis (a) The mass flow rates are

$$\dot{m}_h = \rho_h \dot{V}_h = (994.8 \text{ kg/m}^3)(0.0025/60 \text{ m}^3\text{/s}) = 0.04145 \text{ kg/s}$$

$$\dot{m}_c = \rho_c \dot{V}_c = (998.6 \text{ kg/m}^3)(0.0045/60 \text{ m}^3\text{/s}) = 0.07490 \text{ kg/s}$$

The rates of heat transfer from the hot water and to the cold water are

$$\dot{Q}_h = [\dot{m}c_p(T_{in} - T_{out})]_\text{h} = (0.04145 \text{ kg/s})(4178 \text{ kJ/kg.°C})(38.9\text{°C} - 27.0\text{°C}) = 2061 \text{ W}$$

$$\dot{Q}_c = [\dot{m}c_p(T_{out} - T_{in})]_\text{c} = (0.07490 \text{ kg/s})(4184 \text{ kJ/kg.°C})(19.8\text{°C} - 14.3\text{°C}) = \mathbf{1724 \text{ W}}$$

(b) The logarithmic mean temperature difference and the overall heat transfer coefficient are

$$\Delta T_1 = T_{h,in} - T_{c,out} = 38.9\text{°C} - 19.8\text{°C} = 19.1\text{°C}$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 27.0\text{°C} - 14.3\text{°C} = 12.7\text{°C}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)} = \frac{19.1 - 12.7}{\ln\left(\dfrac{19.1}{12.7}\right)} = 15.68\text{°C}$$

$$U = \frac{\dot{Q}_{hc,m}}{A\Delta T_{lm}} = \frac{(1724 + 2061)/2 \text{ W}}{(0.0400 \text{ m}^2)(15.68\text{°C})} = \mathbf{3017 \text{ W/m}^2 \cdot \text{C}}$$

Note that we used the average of two heat transfer rates in calculations.

(c) The fraction of heat loss and the heat transfer efficiency are

$$f_{loss} = \frac{\dot{Q}_h - \dot{Q}_c}{\dot{Q}_h} = \frac{2061 - 1724}{2061} = 0.164 = \mathbf{16.4\%}$$

$$\eta = \frac{\dot{Q}_c}{\dot{Q}_h} = \frac{1724}{2061} = 0.836 = \mathbf{83.6\%}$$

(d) The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (0.04145 \text{ kg/s})(4178 \text{ kJ/kg.°C}) = 173.2 \text{ W/°C}$$

$$C_c = \dot{m}_c c_{pc} = (0.07490 \text{ kg/s})(4184 \text{ kJ/kg.°C}) = 313.4 \text{ W/°C}$$

Therefore

$$C_{\min} = C_h = 173.2 \text{ W/°C}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate becomes

$$\dot{Q}_{\max} = C_{\min}(T_{h,in} - T_{c,in}) = (173.2 \text{ W/°C})(38.9\text{°C} - 14.3\text{°C}) = 4261 \text{ W}$$

The effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{(1724 + 2061)/2 \text{ kW}}{4261 \text{ kW}} = 0.444 = \mathbf{44.4\%}$$

One again we used the average heat transfer rate. We could have used the smaller or greater heat transfer rates in calculations. The NTU of the heat exchanger is determined from

$$NTU = \frac{UA}{C_{\min}} = \frac{(3017 \text{ W/m}^2 \cdot \text{C})(0.0400 \text{ m}^2)}{173.2 \text{ W/°C}} = \mathbf{0.697}$$

Fundamentals of Engineering (FE) Exam Problems

11-166 Saturated water vapor at 40°C is to be condensed as it flows through the tubes of an air-cooled condenser at a rate of 0.2 kg/s. The condensate leaves the tubes as a saturated liquid at 40°C. The rate of heat transfer to air is

(a) 34 kJ/s (b) 268 kJ/s (c) 453 kJ/s (d) 481 kJ/s (e) 515 kJ/s

Answer (d) 481 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T1=40 [C]
m_dot=0.2 [kg/s]
h_f=ENTHALPY(Steam_IAPWS,T=T1,x=0)
h_g=ENTHALPY(Steam_IAPWS,T=T1,x=1)
h_fg=h_g-h_f
Q_dot=m_dot*h_fg
"Wrong Solutions:"
W1_Q=m_dot*h_f  "Using hf"
W2_Q=m_dot*h_g "Using hg"
W3_Q=h_fg "not using mass flow rate"
W4_Q=m_dot*(h_f+h_g) "Adding hf and hg"
```

11-167 Consider a double-pipe heat exchanger with a tube diameter of 10 cm and negligible tube thickness. The total thermal resistance of the heat exchanger was calculated to be 0.025 ºC/W when it was first constructed. After some prolonged use, fouling occurs at both the inner and outer surfaces with the fouling factors 0.00045 $m^2 \cdot$ºC/W and 0.00015 $m^2 \cdot$ºC/W, respectively. The percentage decrease in the rate of heat transfer in this heat exchanger due to fouling is

(a) 2.3% (b) 6.8% (c) 7.1% (d) 7.6% (e) 8.5%

Answer (c) 7.1%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.10 [m]
R_old=0.025 [C/W]
R_f_i=0.00045 [m^2-C/W]
R_f_o=0.00015 [m^2-C/W]
L=1 [m] "Consider a unit length"
A=pi*D*L
R_fouling=R_f_i/A+R_f_o/A
R_new=R_old+R_fouling
U_old=1/(R_old*A)
U_new=1/(R_new*A)
PercentDecrease=(U_old-U_new)/U_old*Convert(, %)
"Some Wrong Solutions with Common Mistakes"
W1_PercentDecrease=R_fouling/R_old*Convert(, %) "Comparing fouling resistance to old resistance"
W2_R_fouling=R_f_i+R_f_o "Treating fouling factors as fouling resistances"
W2_R_new=R_old+W2_R_fouling
W2_U_new=1/(W2_R_new*A)
W2_PercentDecrease=(U_old-W2_U_new)/U_old*Convert(, %)
```

11-168 Hot water coming from the engine is to be cooled by ambient air in a car radiator. The aluminum tubes in which the water flows have a diameter of 4 cm and negligible thickness. Fins are attached on the outer surface of the tubes in order to increase the heat transfer surface area on the air side. The heat transfer coefficients on the inner and outer surfaces are 2000 and 150 $W/m^2 \cdot {}^oC$, respectively. If the effective surface area on the finned side is 10 times the inner surface area, the overall heat transfer coefficient of this heat exchanger based on the inner surface area is

(a) 150 $W/m^2 \cdot {}^oC$ (b) 857 $W/m^2 \cdot {}^oC$ (c) 1075 $W/m^2 \cdot {}^oC$ (d) 2000 $W/m^2 \cdot {}^oC$ (e) 2150 $W/m^2 \cdot {}^oC$

Answer (b) 857 $W/m^2 \cdot {}^oC$

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.04 [m]
h_i=2000 [W/m^2-C]
h_o=150 [W/m^2-C]
A_i=1 [m^2]
A_o=10 [m^2]
1/(U_i*A_i)=1/(h_i*A_i)+1/(h_o*A_o) "Wall resistance is negligible"

"Some Wrong Solutions with Common Mistakes"
W1_U_i=h_i "Using h_i as the answer"
W2_U_o=h_o "Using h_o as the answer"
W3_U_o=1/2*(h_i+h_o) "Using the average of h_i and h_o as the answer"
```

11-169 A heat exchanger is used to heat cold water entering at 8°C at a rate of 1.2 kg/s by hot air entering at 90°C at rate of 2.5 kg/s. The highest rate of heat transfer in the heat exchanger is

(a) 205 kW (b) 411 kW (c) 311 kW (d) 114 kW (e) 78 kW

Answer (a) 205 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
cp_c=4.18 [kJ/kg-C]
cp_h=1.0 [kJ/kg-C]
Tc_in=8 [C]
Th_in=90 [C]
m_c=1.2 [kg/s]
m_h=2.5 [kg/s]
"From Q_max relation, Q_max=C_min(Th,in-Tc,in)"
Cc=m_c*cp_c
Ch=m_h*cp_h
C_min=min(Cc, Ch)
Q_max=C_min*(Th_in-Tc_in)

"Some Wrong Solutions with Common Mistakes:"
C_max=max(Cc, Ch)
W1Q_max=C_max*(Th_in-Tc_in) "Using Cmax"
```

11-170 Cold water (c_p = 4.18 kJ/kg·ºC) enters a heat exchanger at 15ºC at a rate of 0.5 kg/s where it is heated by hot air (c_p = 1.0 kJ/kg·ºC) that enters the heat exchanger at 50ºC at a rate of 1.8 kg/s. The maximum possible heat transfer rate in this heat exchanger is

(a) 51.1 kW (b) 63.0 kW (c) 66.8 kW (d) 73.2 kW (e) 80.0 kW

Answer (b) 63.0 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_c_in=15 [C]
m_dot_c=0.5 [kg/s]
c_p_c=4.18 [kJ/kg-C]
T_h_in=50 [C]
m_dot_h=1.8 [kg/s]
c_p_h=1.0 [kJ/kg-C]
C_c=m_dot_c*c_p_c
C_h=m_dot_h*c_p_h
C_min=min(C_c, C_h)
Q_dot_max=C_min*(T_h_in-T_c_in)

"Some Wrong Solutions with Common Mistakes"
W1_C_min=C_c "Using the greater heat capacity in the equation"
W1_Q_dot_max=W1_C_min*(T_h_in-T_c_in)
```

11-171 Hot oil (c_p = 2.1 kJ/kg·°C) at 110°C and 8 kg/s is to be cooled in a heat exchanger by cold water (c_p = 4.18 kJ/kg·°C) entering at 10°C and at a rate of 2 kg/s. The lowest temperature that oil can be cooled in this heat exchanger is

(a) 10.0°C (b) 33.5°C (c) 46.1°C (d) 60.2°C (e) 71.4°C

Answer (d) 60.2°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
cp_c=4.18 [kJ/kg-C]
cp_h=2.1 [kJ/kg-C]
Tc_in=10 [C]
Th_in=110 [C]
m_c=2 [kg/s]
m_h=8 [kg/s]
"From Q_max relation, Q_max=C_min(Th,in-Tc,in)"
Cc=m_c*cp_c
Ch=m_h*cp_h
C_min=min(Cc, Ch)
Q_max=C_min*(Th_in-Tc_in)
Q_max=Ch*(Th_in-Th_out)

“Some Wrong Solutions with Common Mistakes:”
C_max=max(Cc, Ch)
W1Q_max=C_max*(Th_in-Tc_in) “Using Cmax”
W1Q_max=Ch*(Th_in-W1Th_out)
```

11-172 Cold water (c_p = 4.18 kJ/kg·ºC) enters a counter-flow heat exchanger at 18ºC at a rate of 0.7 kg/s where it is heated by hot air (c_p = 1.0 kJ/kg·ºC) that enters the heat exchanger at 50ºC at a rate of 1.6 kg/s and leaves at 25ºC. The maximum possible outlet temperature of the cold water is

(a) 25.0ºC (b) 32.0ºC (c) 35.5ºC (d) 39.7ºC (e) 50.0ºC

Answer (c) 35.5ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_c_in=18 [C]
m_dot_c=0.7 [kg/s]
c_p_c=4.18 [kJ/kg-C]
T_h_in=50 [C]
T_h_out=25 [C]
m_dot_h=1.6 [kg/s]
c_p_h=1.0 [kJ/kg-C]
C_c=m_dot_c*c_p_c
C_h=m_dot_h*c_p_h
C_min=min(C_c, C_h)
Q_dot_max=C_min*(T_h_in-T_c_in)
Q_dot_max=C_c*(T_c_out_max-T_c_in)
"Some Wrong Solutions with Common Mistakes"
W1_C_min=C_c "Using the greater heat capacity in the equation"
W1_Q_dot_max=W1_C_min*(T_h_in-T_c_in)
W1_Q_dot_max=C_c*(W1_T_c_out_max-T_c_in)
W2_T_c_out_max=T_h_in "Using T_h_in as the answer"
W3_T_c_out_max=T_h_out "Using T_h_in as the answer"
```

11-173 A heat exchanger is used to condense steam coming off the turbine of a steam power plant by cold water from a nearby lake. The cold water (c_p = 4.18 kJ/kg·ºC) enters the condenser at 16ºC at a rate of 20 kg/s and leaves at 25ºC while the steam condenses at 45ºC. The condenser is not insulated and it is estimated that heat at a rate of 8 kW is lost from the condenser to the surrounding air. The rate at which the steam condenses is

(a) 0.282 kg/s (b) 0.290 kg/s (c) 0.305 kg/s (d) 0.314 kg/s (e) 0.318 kg/s

Answer (e) 0.318 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_c_in=16 [C]
T_c_out=25 [C]
m_dot_c=20 [kg/s]
c_p_c=4.18 [kJ/kg-C]
T_h=45 [C]
Q_dot_lost=8 [kW]
Q_dot_c=m_dot_c*c_p_c*(T_c_out-T_c_in) "Heat picked up by the cold fluid"
Q_dot_h=Q_dot_c+Q_dot_lost "Heat given up by the hot fluid"
h_fg=2395 [kJ/kg] "Table A-9"
m_dot_cond=Q_dot_h/h_fg
"Some Wrong Solutions with Common Mistakes"
W1_m_dot_cond=Q_dot_c/h_fg "Ignoring heat loss from the heat exchanger"
```

11-174 An air handler is a large unmixed heat exchanger used for comfort control in large buildings. In one such application, chilled water (c_p = 4.2 kJ/kg·K) enters an air handler at 5°C and leaves at 12°C with a flow rate of 1000 kg/h. This cold water cools 5000 kg/h of air (c_p = 1.0 kJ/kg·K) which enters the air handler at 25°C. If these streams are in counter-flow and the water stream conditions remain fixed, the minimum temperature at the air outlet is

(a) 5°C (b) 12°C (c) 19°C (d) 22°C (e) 25°C

Answer (c) 19°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
cp_c=4.2 [kJ/kg-K]
T_c_in=5 [C]
T_c_out=12 [C]
m_dot_c=1000/3600 "[kg/s]"
m_dot_h=5000/3600 "[kg/s]"
cp_h=1.0 [kJ/kg-K]
T_h_in=25 [C]
Q_dot=m_dot_c*cp_c*(T_c_out-T_c_in)
Q_dot=m_dot_h*cp_h*(T_h_in-T_h_out)
```

11-175 An air handler is a large unmixed heat exchanger used for comfort control in large buildings. In one such application, chilled water (c_p = 4.2 kJ/kg·K) enters an air handler at 5°C and leaves at 12°C with a flow rate of 1000 kg/hr. This cold water cools air (c_p = 1.0 kJ/kg·K) from 25°C to 15°C. The rate of heat transfer between the two streams is

(a) 8.2 kW (b) 23.7 kW (c) 33.8 kW (d) 44.8 kW (e) 52.8 kW

Answer (a) 8.2 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
cp_c=4.2 [kJ/kg-K]
T_c_in=5 [C]
T_c_out=12 [C]
m_dot_c=1000/3600 "[kg/s]"
cp_h=1.0 [kJ/kg-K]
T_h_in=25 [C]
T_h_out=15 [C]
Q_dot=m_dot_c*cp_c*(T_c_out-T_c_in)
```

11-176 In a parallel-flow, liquid-to-liquid heat exchanger, the inlet and outlet temperatures of the hot fluid are 150ºC and 90ºC while that of the cold fluid are 30ºC and 70ºC, respectively. For the same overall heat transfer coefficient, the percentage decrease in the surface area of the heat exchanger if counter-flow arrangement is used is

(a) 3.9% (b) 9.7% (c) 14.5% (d) 19.7% (e) 24.6%

Answer (e) 24.6%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_h_in=150 [C]
T_h_out=90 [C]
T_c_in=30 [C]
T_c_out=70 [C]
"Parallel flow arrangement"
DELTAT_1_p=T_h_in-T_c_in
DELTAT_2_p=T_h_out-T_c_out
DELTAT_lm_p=(DELTAT_1_p-DELTAT_2_p)/ln(DELTAT_1_p/DELTAT_2_p)
"Counter flow arrangement"
DELTAT_1_c=T_h_in-T_c_out
DELTAT_2_c=T_h_out-T_c_in
DELTAT_lm_c=(DELTAT_1_c-DELTAT_2_c)/ln(DELTAT_1_c/DELTAT_2_c)
PercentDecrease=(DELTAT_lm_c-DELTAT_lm_p)/DELTAT_lm_p*Convert(, %)
"From Q_dot = U*A_s *DELTAT_lm, for the same Q_dot and U, DELTAT_lm and A_s are inversely proportional."

"Some Wrong Solutions with Common Mistakes"
W_PercentDecrease=(DELTAT_lm_c-DELTAT_lm_p)/DELTAT_lm_c*Convert(, %) "Dividing the difference by DELTAT_lm_c "
```

11-177 A counter-flow heat exchanger is used to cool oil (c_p = 2.20 kJ/kg·ºC) from 110ºC to 85ºC at a rate of 0.75 kg/s by cold water (c_p = 4.18 kJ/kg·ºC) that enters the heat exchanger at 20ºC at a rate of 0.6 kg/s. If the overall heat transfer coefficient is 800 W/m^2·ºC, the heat transfer area of the heat exchanger is

(a) 0.745 m^2 (b) 0.760 m^2 (c) 0.775 m^2 (d) 0.790 m^2 (e) 0.805 m^2

Answer (a) 0.745 m^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_h_in=110 [C]
T_h_out=85 [C]
m_dot_h=0.75 [kg/s]
c_p_h=2.20 [kJ/kg-C]
T_c_in=20 [C]
m_dot_c=0.6 [kg/s]
c_p_c=4.18 [kJ/kg-C]
U=0.800 [kW/m^2-C]
Q_dot=m_dot_h*c_p_h*(T_h_in-T_h_out)
Q_dot=m_dot_c*c_p_c*(T_c_out-T_c_in)
DELTAT_1=T_h_in-T_c_out
DELTAT_2=T_h_out-T_c_in
DELTAT_lm=(DELTAT_1-DELTAT_2)/ln(DELTAT_1/DELTAT_2)
Q_dot=U*A_s*DELTAT_lm
```

11-178 The radiator in an automobile is a cross-flow heat exchanger (UA_s = 10 kW/K) that uses air (c_p = 1.00 kJ/kg·K) to cool the engine coolant fluid (c_p = 4.00 kJ/kg·K). The engine fan draws 30^{o}C air through this radiator at a rate of 10 kg/s while the coolant pump circulates the engine coolant at a rate of 5 kg/s. The coolant enters this radiator at 80^{o}C. Under these conditions, what is the number of transfer units (NTU) of this radiator?

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

Answer (c) 3

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
UA=30 [kW/K]
m_dot_a=10 [kg/s]
c_p_a=1.0 [kJ/kg-K]
m_dot_c=5 [kg/s]
c_p_c=4.0 [kJ/kg-K]
C_a=m_dot_a*c_p_a
C_c=m_dot_c*c_p_c
C_min=C_a
NTU=UA/C_min
```

11-179 In a parallel-flow, water-to-water heat exchanger, the hot water enters at 75ºC at a rate of 1.2 kg/s and cold water enters at 20ºC at a rate of 0.9 kg/s. The overall heat transfer coefficient and the surface area for this heat exchanger are 750 W/m^2·ºC and 6.4 m^2, respectively. The specific heat for both the hot and cold fluid may be taken to be 4.18 kJ/kg·ºC. For the same overall heat transfer coefficient and the surface area, the increase in the effectiveness of this heat exchanger if counter-flow arrangement is used is

(a) 0.09 (b) 0.11 (c) 0.14 (d) 0.17 (e) 0.19

Answer (a) 0.09

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_h_in=75 [C]
m_dot_h=1.2 [kg/s]
T_c_in=20 [C]
m_dot_c=0.9 [kg/s]
c_p=4.18 [kJ/kg-C]
U=0.750 [kW/m^2-C]
A_s=6.4 [m^2]
C_h=m_dot_h*c_p
C_c=m_dot_c*c_p
C_min=min(C_c, C_h)
C_max=max(C_c, C_h)
c=C_min/C_max
NTU=(U*A_s)/C_min
epsilon_p=(1-exp((-NTU)*(1+c)))/(1+c)
epsilon_c=(1-exp((-NTU)*(1-c)))/(1-c*exp((-NTU)*(1-c)))
Increase_epsilon=epsilon_c-epsilon_p
```

11-180 In a parallel-flow heat exchanger, the NTU is calculated to be 2.5. The lowest possible effectiveness for this heat exchanger is

(a) 10% (b) 27% (c) 41% (d) 50% (e) 92%

Answer (d) 50%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
NTU=2.5
c=1 "The effectiveness is lowest when c = 1"
epsilon=(1-exp((-NTU)*(1+c)))/(1+c)

"Some Wrong Solutions with Common Mistakes"
W_epsilon=1-exp(-NTU) "Finding maximum effectiveness when c=0"
```

11-181 Cold water (c_p = 4.18 kJ/kg·ºC) enters a counter-flow heat exchanger at 10ºC at a rate of 0.35 kg/s where it is heated by hot air (c_p = 1.0 kJ/kg·ºC) that enters the heat exchanger at 50ºC at a rate of 1.9 kg/s and leaves at 25ºC. The effectiveness of this heat exchanger is

(a) 0.50 (b) 0.63 (c) 0.72 (d) 0.81 (e) 0.89

Answer (d) 0.81

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_c_in=10 [C]
m_dot_c=0.35 [kg/s]
c_p_c=4.18 [kJ/kg-C]
T_h_in=50 [C]
T_h_out=25 [C]
m_dot_h=1.9 [kg/s]
c_p_h=1.0 [kJ/kg-C]
C_c=m_dot_c*c_p_c
C_h=m_dot_h*c_p_h
C_min=min(C_c, C_h)
Q_dot_max=C_min*(T_h_in-T_c_in)
Q_dot=m_dot_h*c_p_h*(T_h_in-T_h_out)
epsilon=Q_dot/Q_dot_max

"Some Wrong Solutions with Common Mistakes"
W1_C_min=C_h "Using the greater heat capacity in the equation"
W1_Q_dot_max=W1_C_min*(T_h_in-T_c_in)
W1_epsilon=Q_dot/W1_Q_dot_max
```

11-182 Steam is to be condensed on the shell side of a 2-shell-passes and 8-tube-passes condenser, with 20 tubes in each pass. Cooling water enters the tubes at a rate of 2 kg/s. If the heat transfer area is 14 m^2 and the overall heat transfer coefficient is 1800 W/m^2·°C, the effectiveness of this condenser is

(a) 0.70 (b) 0.80 (c) 0.90 (d) 0.95 (e) 1.0

Answer (d) 0.95

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
cp_c=4.18 [kJ/kg-C]
m_c=2 [kg/s]
A=14
U=1.8 [kW/m^2-K]
"From NTU and Effectivenss relations for counterflow HX:"
C_min=m_c*cp_c
NTU=U*A/C_min
Eff=1-Exp(-NTU)
```

11-183 Water is boiled at 150ºC in a boiler by hot exhaust gases (c_p = 1.05 kJ/kg·ºC) that enter the boiler at 400ºC at a rate of 0.4 kg/s and leaves at 200ºC. The surface area of the heat exchanger is 0.64 m^2. The overall heat transfer coefficient of this heat exchanger is

(a) 940 W/m^2·ºC (b) 1056 W/m^2·ºC (c) 1145 W/m^2·ºC (d) 1230 W/m^2·ºC (e) 1393 W/m^2·ºC

Answer (b) 1056 W/m^2·ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_w=150 [C]
T_h_in=400 [C]
T_h_out=200 [C]
m_dot_h=0.4 [kg/s]
c_p_h=1.05 [kJ/kg-C]
A_s=0.64 [m^2]
C_h=m_dot_h*c_p_h
C_min=C_h
Q_dot_max=C_min*(T_h_in-T_w)
Q_dot=C_h*(T_h_in-T_h_out)
epsilon=Q_dot/Q_dot_max
NTU=-ln(1-epsilon)
U=(NTU*C_min)/A_s
```

11-184 An air-cooled condenser is used to condense isobutane in a binary geothermal power plant. The isobutane is condensed at 85ºC by air (c_p = 1.0 kJ/kg·ºC) that enters at 22ºC at a rate of 18 kg/s. The overall heat transfer coefficient and the surface area for this heat exchanger are 2.4 kW/m^2·ºC and 1.25 m^2, respectively. The outlet temperature of the air is

(a) 45.4ºC (b) 40.9ºC (c) 37.5ºC (d) 34.2ºC (e) 31.7ºC

Answer (e) 31.7ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_h=85 [C]
T_c_in=22 [C]
m_dot_c=18 [kg/s]
c_p_c=1.0 [kJ/kg-C]
U=2.4 [kW/m^2-C]
A_s=1.25 [m^2]
C_c=m_dot_c*c_p_c
C_min =C_c
NTU=(U*A_s)/C_min
epsilon=1-exp(-NTU)
Q_dot_max=C_min*(T_h-T_c_in)
Q_dot=epsilon*Q_dot_max
Q_dot=m_dot_c*c_p_c*(T_c_out-T_c_in)
```

11-185 . . . 11-191 Design and Essay Problems

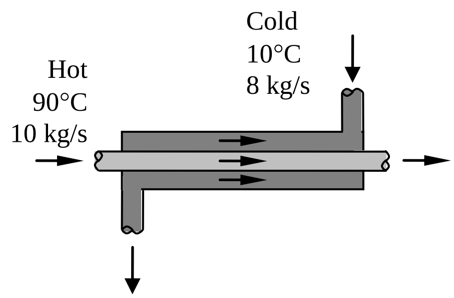
11-191 A counter flow double-pipe heat exchanger is used for cooling a liquid stream by a coolant. The rate of heat transfer and the outlet temperatures of both fluids are to be determined. Also, a replacement proposal is to be analyzed.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There is no fouling.

Properties The specific heats of hot and cold fluids are given to be 3.15 and 4.2 kJ/kg.°C, respectively.

Analysis (a) The overall heat transfer coefficient is

$$U=\frac{600}{\dfrac{1}{\dot m_c^{0.8}}+\dfrac{2}{\dot m_h^{0.8}}}=\frac{600}{\dfrac{1}{8^{0.8}}+\dfrac{2}{10^{0.8}}}=1185\ \text{W/m}^2.\text{K}$$



The rate of heat transfer may be expressed as

$$\dot Q=\dot m_c c_c(T_{c,out}-T_{c,in})=(8)(4200)(T_{c,out}-10) \qquad (1)$$

$$\dot Q=\dot m_h c_h(T_{h,in}-T_{h,out})=(10)(3150)(90-T_{h,out}) \qquad (2)$$

It may also be expressed using the logarithmic mean temperature difference as

$$\dot Q=UA\Delta T_{lm}=UA\frac{\Delta T_1-\Delta T_2}{\ln(\Delta T_1/\Delta T_2)}=(1185)(9)\frac{(90-T_c)-(T_h-10)}{\ln\left(\dfrac{90-T_c}{T_h-10}\right)} \qquad (3)$$

We have three equations with three unknowns, solving an equation solver such as EES, we obtain

$$\dot Q=\mathbf{6.42\times10^5\ W},\quad T_{c,out}=\mathbf{29.1°C},\quad T_{h,out}=\mathbf{69.6°C}$$

(b) The overall heat transfer coefficient for each unit is

$$U=\frac{600}{\dfrac{1}{\dot m_c^{0.8}}+\dfrac{2}{\dot m_h^{0.8}}}=\frac{600}{\dfrac{1}{4^{0.8}}+\dfrac{2}{5^{0.8}}}=680.5\ \text{W/m}^2.\text{K}$$

Then

$$\dot Q=\dot m_c c_c(T_{c,out}-T_{c,in})=(2\times4)(4200)(T_{c,out}-10) \qquad (1)$$

$$\dot Q=\dot m_h c_h(T_{h,in}-T_{h,out})=(2\times5)(3150)(90-T_{h,out}) \qquad (2)$$

$$\dot Q=UA\Delta T_{lm}=UA\frac{\Delta T_1-\Delta T_2}{\ln(\Delta T_1/\Delta T_2)}=(680.5)(2\times5)\frac{(90-T_c)-(T_h-10)}{\ln\left(\dfrac{90-T_c}{T_h-10}\right)} \qquad (3)$$

Once again, we have three equations with three unknowns, solving an equation solver such as EES, we obtain

$$\dot Q=\mathbf{4.5\times10^5\ W},\quad T_{c,out}=\mathbf{23.4°C},\quad T_{h,out}=\mathbf{75.7°C}$$

Discussion Despite a higher heat transfer area, the new heat transfer is about 30% lower. This is due to much lower *U*, because of the halved flow rates. So, the vendor's recommendation is not acceptable. The vendor's unit will do the job provided that they are connected in series. Then the two units will have the same *U* as in the existing unit.